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J. F. KELLY, Editor

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SATURDAY, MARCH 15, 1930

"STEEL PLANT SWITCHING AND FEEDER SYSTEM EQUIPMENT"

The Short Circuit Study
Selection and Application of the Oil Circuit Breaker
Relaying the Bus and Feeder System

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8:00 P. M. LECTURE

"AN EVENING IN THE RESEARCH LABORATORIES OF THE GENERAL ELECTRIC COMPANY"

By L. A. HAWKINS, Executive Engineer Research Laboratories

The Development of the Cathode Ray
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SATURDAY, MARCH 1, 1930

"MAGNETIC CLUTCHES IN INDUSTRY"

By H. E. HODGSON, Cutler Hammer Inc., Milwaukee, Wis.

SATURDAY, APRIL 5, 1930

"GENERAL IDEAS AND DISCUSSIONS"

Every Active Member is invited to prepare a short paper on a subject pertaining to the application of electricity to the Iron and Steel Industry. Mr. John C. Reed, Electrical Engineer, Bethlehem Steel Co., Steelton, Pa., will present a paper on "Some Applications of Fusion Welding to the Maintenance of Electrical Equipment in the Steel Industry."

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"SYMPOSIUM ON CONTROL EQUIPMENT"

By Cutler Hammer, Inc., Electric Controller & Mfg. Co., Clark Controller Company, General Electric Co., Westinghouse Electric & Mfg. Co., Rowan Controller Co., Union Electric Mfg. Co.

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SATURDAY, MARCH 29, 1930

"INTERNAL COMBUSTION TYPE LOCOMOTIVES FOR STEEL PLANT TRANSPORTATION"

By A. H. CANDY, Section Head of Diesel Electric and Oil Electric Locomotives, Westinghouse Electric & Mfg. Co.

A SURVEY
of the
IRON AND STEEL INDUSTRY

The IRON & STEEL ENGINEER, a periodical rendering an exclusive engineering service to the Steel Industry, has undertaken to present to the General Sales Manager, Advertising Manager, Advertising Agencies and Sales Engineers of the United States, who service this vast Industry a number of the essential factors, which must be taken into consideration when planning sales campaigns, advertising programs, distribution of catalogues, or trade literature, and when calling on the prospective customers.

Many volumes are written on the processes and methods used in production of Iron and Steel, but not a great deal of literature is available that really indicates the physical structure of a plant, the apparatus used, its location, also its relation to the finished product of the plant.

It was suggested, by a number of Steel Plant Executives, that the preparation of a small pamphlet, outlining the various departments, their physical structure, the apparatus used, how a steel plant is organized and functions, might be of assistance to manufacturers and their representatives.

With this end in view, the IRON & STEEL ENGINEER, through the personnel of the A. I. & S. E. E., has compiled information and data covering the apparatus and materials involved.

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Tapered Roller Bearings and Their Applicability to Steel Mill Equipment

By F. WALDORF*

It probably can be said, without immediate danger of contradiction or criticism, that no branch of steel mill engineering in all of its ramifications has lately incited so much interest or curiosity as that phase which deals with the application of anti-friction bearings to roll necks.

It is the one subject that deserves, and is recommended for, the earnest consideration of the steel mill engineer. A careful investigation of the possibilities it offers will not only supplement his knowledge of the art of rolling steel, but it will enable him to find ways and means of effecting numerous economies, which, if properly instituted, will do much to increase the earnings his company is in existence to enjoy.

Half a decade is not a long while when viewed in the light of the speed at which we are traveling today. Yet, in less than that period of time, we have seen, or heard of, probably a hundred rolling mills being built around roller bearings.

That the success so far enjoyed with them has been extremely gratifying is a matter of common knowledge, for, with the exception of two high hot sheet mills, blooming and plate mills, one rarely hears of a new installation going in on plain bearings.

When considered only from the standpoint of maintaining production, roller bearings have long ago justified their adoption through that utility alone. Mills pay their way only as they continue in operation, and it is only natural that any innovation which will insure practically uninterrupted production should receive every consideration and attention.

You are all familiar with the items of cost that are added to rolling charges to cover plain bearing up-keep, their lubrication and renewals. You know also the magnitudes of delay charges incident to shut downs that are traced to any of those items. It therefore must appear to you as folly to continue to contend with such conditions when, by so much as applying roller bearings to the roll necks, a mill can be converted into one of the most economical pieces of equipment it is possible to construct.

The industry, so far as strip rolling is concerned, has for several years enjoyed prosperity that can truthfully be laid at the door of the roller bearing builder. We do not make so bold as to infer that it would not have enjoyed some prosperity if the rolling had been done on plain bearings. They certainly would have answered for a time so far as maintenance of gauge was involved, and, with lubrication bordering on the ideal, so far as power consumption was concerned.

But plain bearings wear rapidly. Foreign matter finds its way to the necks, which in time begin to score, and ideal lubrication becomes a difficult matter to maintain. Power consumption increases as a result, while finish and gauge of the strip suffer so materially as to need continual watching.

Roller bearings, on the other hand, wear slowly, almost imperceptibly. They are almost frictionless. Lubrication and maintenance are items that need so little attention as to be already classed with the necessary evils that accompany all endeavor that mankind undertakes.

And above all else, gauges are attained, and maintained with uniformity and precision, which, a few years ago, would have been scoffed at and considered as absolutely impossible of realization.

That the demand for bearings of special characteristics maintains the lead over the art of producing them cannot be denied. This condition will undoubtedly continue since, to date, there has been very little promise that such a thing as standardizing bearings will ever be possible, save, perhaps, such sizes as are used where similarity, or duplication, in the design of equipment is possible. However, if the bearing manufacturer is given the co-operation he needs, he is able today to produce anti-frictions that will meet the most exacting requirements.

The most popular types of bearings in use today are built around the straight, the barrel shaped and the tapered roller principles. Each type however differs distinctly from the others, and it was early learned that it was impracticable to attempt a design to meet definite given over all dimensions rather than to hold to the proportions which are inherent in the respective types mentioned.

By far the most difficult problem the mill engineers and bearing builders have to solve is that of determining the loads that the bearing must sustain. Speeds can be taken with stop watches, or recorded with tachometers, but the method of accurately determining the rolling loads is, as yet, far from being satisfactorily established.

Hydraulic cylinders have been used to some extent in this country. Dr. Puppe and his associates have used it in some hundred or more tests they conducted in Germany. In an article which appeared in Stahl und Eisen in 1911 he describes his cylinder and places particular emphasis on the care that was taken when the cylinder was calibrated in order that extreme accuracy should be attained. He went even so far as to determine factors which would enable him to make corrections for plunger and packing friction under varying pressures, and he further published separate factors which made provisions for any eccentricity of loading on the plunger in the event such eccentricity existed.

What appears to be the only real objection to the hydraulic cylinder is the fact that it is bulky, and very heavy equipment to transport and handle. There is also the objection that one size of cylinder

*District Manager, Steel Mill Division, Timken Roller Bearing Company, Canton, Ohio.

will not answer for every test, because the space available is not always adequate to accommodate it.

It therefore appears certain that several sizes of cylinders should be made available, and certainly only such a size should be used which will allow sufficient travel of the needle over the recording chart to permit of accurately reading the range in pressures reached during the rolling operation.

It is the speaker's belief that if enough range in sizes of hydraulic cylinders is made available, all objection to its use as a medium for ascertaining rolling loads will disappear and that it will eventually be the only method used if extreme accuracy and reliability is to be sought.

Leaving the general to take up the particular, the progress made in anti-friction bearing applications has been rapid, especially during the past two or three years. How much the development of methods of obtaining accurate data on load characteristics has contributed to its rapidity may be an open question; the fact remains that both the number and variety of mills equipped is steadily increasing. The results are surprisingly uniform with regard to successful operation. To make a long story short, the anti-friction bearing has thoroughly established both its suitability and its advantageous characteristics in steel mill practice. How thoroughly can probably be seen best from a brief outline of what has actually been accomplished.

4 High Cold Mills

During the past three years quite a number of tapered roller bearing applications have been made to varying sizes of 4 high cold strip mills. While it was impractical to prepare slides showing the details of all of the mountings developed for this particular type of mill, it is possible to describe the applications that have been made, and to give some idea of the load conditions encountered in this service. For example, 10 stands of 4 high cold strip mills have been in operation for three years with Timken bearings on the back-up rolls. Six of these mills are in one plant, and four are in another. The bearings are 16-15/16x26½x13 inches wide with a rated capacity of 900,000 lb. per neck at 50 R.P.M. After installation a test was made in one plant to determine the actual load developed when rolling 37¼-inch wide, .10 carbon unannealed sheets from .084" to .073" or a 13.1% reduction. The test showed an actual load of 750,000 lb. per back-up neck at 8 R.P.M. A test made at the other plant showed a load of approximately 975,000 lb. at 12½ R.P.M. The material in this case was unannealed 30-inch strip of .08 to .12 carbon, the reduction being from .083" to .070", or 15.7%. It has also been possible to make a test on another 4 high cold mill using 18x28¾x16-inch bearings with a capacity of 1,500,000 lb. per neck at 20 R.P.M. When rolling 40-inch .12 carbon strip with a reduction from .075" to .054" or approximately 29%, the mill developed a load of 1,204,000 lb. per neck.

4 High Hot Strip Mills

Equally successful operation has been obtained on the back-up rolls of 4 high hot strip mills. In one installation, nine stands using our 23x35½x21¼-inch bearings, which have a capacity of 2,800,000 lb. at 35 R.P.M., have been working for almost one year. In addition there is one installation using our 22x35½x18-inch bearings in one stand of 4 high hot mill, having a capacity per neck of 2,500,000 lbs. at 20 R.P.M. While the opportunity for making tests on these mills has never arisen, the results of a test made for recommendation purposes on another mill will serve to show the magnitude of the load encountered. While rolling 30-inch, .08 to .12 carbon steel sheets with a reduction from .167" to .104" or approximately 38%, the mill developed a load of approximately 1,300,000 lb. per neck, at 80 R.P.M. For this installation 18x30x20¾-inch bearing, having a capacity of 1,800,000 lb. per neck at 80 R.P.M. are now being manufactured.

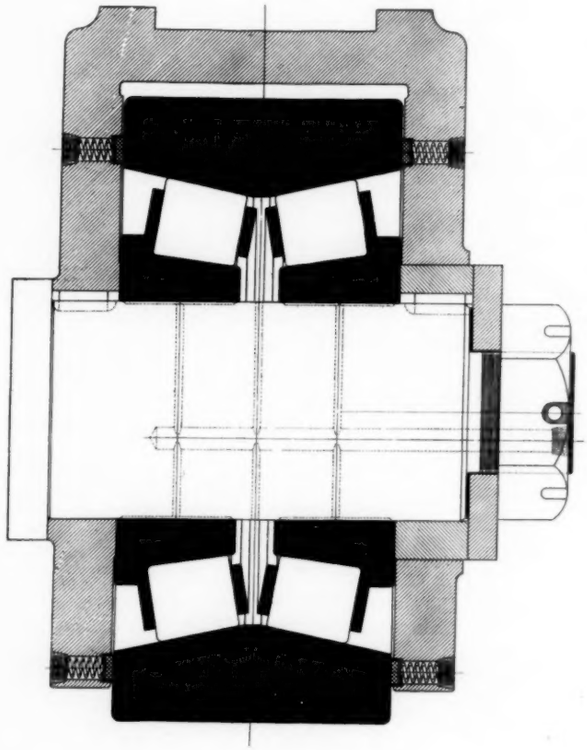


FIG. 1.

A case of particular interest is one in which 7⅝x22x10-inch Timken bearings are being used as the back-up rolls proper on five stands of a specially designed 4 high hot strip mill. Figure 1 gives the details of the arrangement adopted in this installation. Two standard cones are used, together with a double cup made of special steel to give wearing quality while it acts as the back-up roll. The cup section has been made large enough to allow for considerable reduction in diameter due to dressings. Closures are specially designed to assure proper lubrication. The capacity of the roll is 350,000 lb. at 100 R.P.M. In this connection it may be said that two stands of similar design with this same bearing acting as back-up roll have been running for two years without trouble.

In addition to the applications made to back-up rolls, application of Timkens has been made to the working rolls of 4 high hot strip mills, with an equal degree of success.

Cluster Mills

A considerable number of applications have been made to cluster mills of various sizes. In two plants, where mills of this type have been in service for over three years, 12x19½x11¾-inch bearings are used on the back-up rolls. These bearings have a capacity of 450,000 lb. at 50 R.P.M. Another installation uses our 16x25x12½-inch bearings with capacities of 715,000 lb. at 50 R.P.M. each. In the smaller mills, some use 10¼x15¾x10-inch bearings with capacities of 180,000 lb. at 50 R.P.M., and others 7½x13x9¼-inch bearings with capacities of 120,000 lb. at 50 R.P.M.

No pressure tests have been made on cluster mills to compare actual load with rated bearing capacity. However, the results obtained in one case where a load test was made before installation brought out some interesting facts. The mill tested was used to roll sheets of special analysis monel metal, and the average load under various rolling conditions was found to be from 250,000 to 350,000 lb. at 15 R.P.M. The bearings installed were 13x21x11½-inch, with a capacity of 600,000 lb. per neck at 15 R.P.M. The excess capacity was provided to permit of increasing the mill speed should it become desirable. You will be interested in learning that power checks made on light duty operation of this particular mill showed a 60% saving after the installation of the Timken bearings. Even greater saving is expected when the mill is run to full capacity.

2 High Cold Sheet Mills

Because of the present space limitations in the roll housings, progress in applying roller bearings to 2 high cold mills has been rather slow. There are in operation, however, two 18-inch cold mills using 10¼x15¾x10-inch bearings with a capacity of 180,000 lb. per neck at 50 R.P.M. In addition, a test is now being run on a 26-inch, 2 high cold mill using 16x24x12½-inch bearings with a capacity of 586,000 lb. per neck at 50 R.P.M. The results obtained so far in this mill have been very gratifying and encouraging, from every standpoint.

Bar and Rod Mills

A number of bearing applications have been made to bar and rod mills used for rolling different metals, such as copper, aluminum, or steel. Since the applications made to the mills at the plant of the Timken Steel and Tube Company are fairly representative of the general methods used in mounting bearings, they will serve as a basis for a discussion of what has been accomplished so far.

One typical application is that made to our 28-inch bar mill which, in regular practice, breaks down 11"x13" blooms to 5"x5" billets. The complete unit consists of two stands of 3 high 28-inch mill, driven through a 3 high pinion stand and gear reduction unit by a 1500 H.P. induction motor. This mill uses 16x24x12½-inch bearings with a capacity of 586,000 lb. per neck at 50 R.P.M. It started operation in October, 1928, and has been in continuous service to date without any bearing failures. During this time an accurate record of the tonnage rolled has been kept to furnish data on bearing life. The record shows that some of the bearings have rolled as much as 100,000 tons, without any deterioration that is evident to a close inspection. Figure 2 shows the details of the mounting.

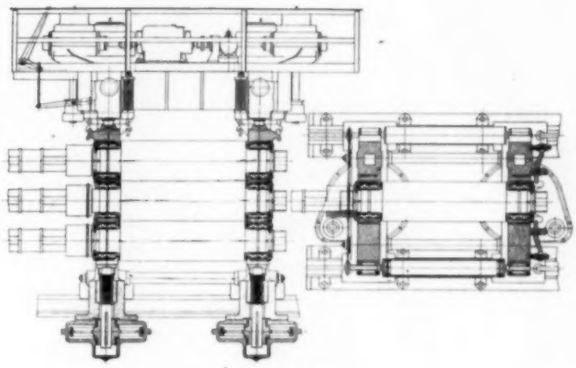


FIG. 2.

One of the interesting features of the arrangement is the provision that has been made to secure proper pass alignment. The middle roll is so fixed in the housing as to have no lateral or vertical adjustment. The bearings in the right hand chuck are fixed on the roll neck, while those in the left hand chuck are allowed to float on the neck to take care of roll expansion. Both of the chucks are rigidly bolted to the housing. Pass alignment is provided for in the design of the top and bottom rolls, as can be clearly seen from the view at the right. On these rolls, the bearings at each end are fixed on the necks. The chucks at one end are free to float in the housings, while those on the opposite end are provided with lugs which engage with specially designed clamps. Manipulation of the nuts on the clamps causes the rolls to move in or out as desired, so that the passes can be lined up. This cut is an exact reproduction of the way the mill was built with the exception of a slight change in the method of locking the bearings on the necks. This had to be changed because of lock breakage due to faulty original design.

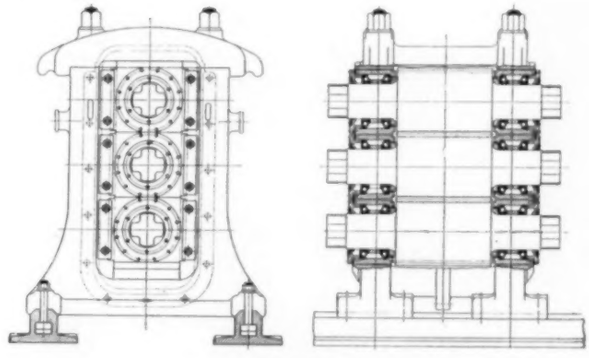


FIG. 3.

Figure 3 shows the general bearing arrangement on the 28-inch 3 high pinion stand. The bearings used here are the same size as those used on the mill rolls, except that only one-half (½) of the bearing capacity per neck is provided. As will be seen, single row bearings separated by a spacer between the cones are used, instead of the usual two row type. In this case, the bearing chucks are positively fixed in the housings, while the bearing cones are given a loose fit on the necks. Clearance is provided between the inner cone and the fillet

ring to take care of both pinion float and any expansion that may arise. The whole set-up you will note is a very simple, but effective, arrangement.

The next mill for consideration is our 22-inch. This consists of three 3 high stands and one 2 high stand, driven through one 3 high pinion stand and a single reduction herringbone gear unit by a 2,000 H.P. direct current motor. This mill, as a rule, takes 5″x5″ billets which it rolls to 3¾″ rounds. It is, however, designed to break down 9″x9½″ blooms into 5″x5″ billets. Figure 4 shows the general ar-

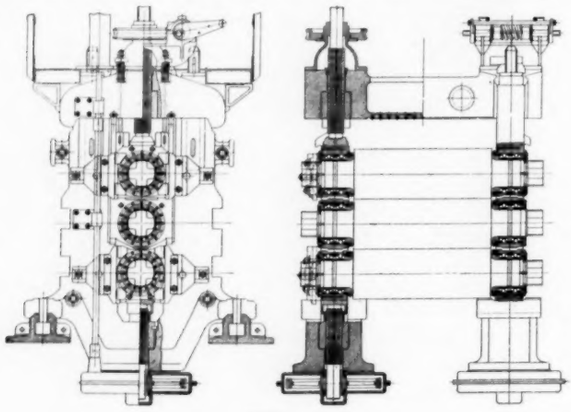


FIG. 4.

rangement of the mill. The bearings used are 12x-17½x9½ inches with a capacity per neck of 156,000 lb. at 100 R.P.M. It is of interest that before this complete installation was made a bearing of the same size was thoroughly tested on one stand of a 22-inch mill, where it remained in operation for two and a half years without any signs of trouble.

The view given in the figure clearly shows the provisions made for securing pass alignment, as well as the method used for fixing the middle roll in the housing. The middle chuck is provided with two upward projecting horns which are wedged under the mill cap, the top roll chuck fitting into the projections instead of into the housing. This arrangement facilitates the changing of rolls with but little loss of time. The same locking arrangement is used as was shown for the 28-inch mill.

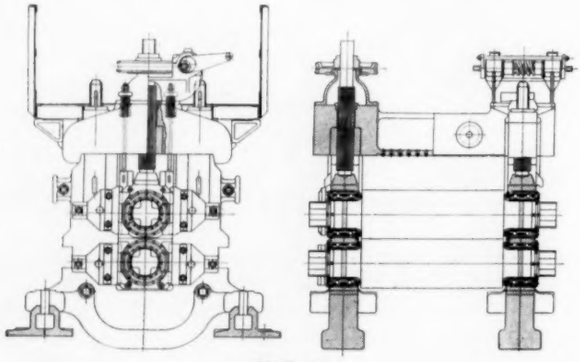


FIG. 5.

Figure 5 shows the mounting on the 2 high 22-inch mill stand, which is built but not yet installed. The general details of the bearing mounting, and method of obtaining pass alignment, are the same as those previously described.

The pinion stand on the 22-inch mill has been in operation for almost three years. The bearings used in it are 13⅝x19x8¾ inches with a capacity per neck of 99,000 lb. at 100 R.P.M. The mounting is the same as that used for the 28-inch pinion stand, except that a 4 row bearing is used.

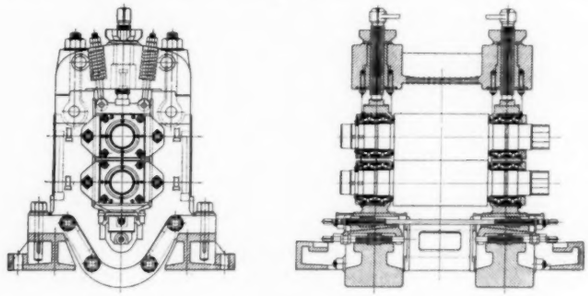


FIG. 6.

Figure 6 shows the general assembly of the bearings on a 2 high 18-inch stand that forms part of a 16 stand continuous merchant mill which rolls 6″x6″ billets to 1⅝″ squares in the first eight passes. In this case, the design is quite different from any of those previously discussed. The principal feature of interest is the arrangement whereby, instead of clamping the bearings on the neck, the same effect is obtained by using bearings with two different cone bores on one end of the roll. On the opposite end the bores are uniform, permitting the cones to float to take care of roll expansion. The bottom roll is fixed in the housing by bolting the chucks to same. The top roll has been rendered moveable laterally by means of a chuck on one end which can be adjusted by means of special bolts. The chuck at the other end is free to float, but is prevented from moving out of the mill by a clamp that is bolted to the housing, but that does not touch the chuck. On the adjustable chuck, the lug holes are elongated to take care of rolls of different diameters.

In the right hand view it is possible to see how the use of bearings with different diameter bores clamps the outer cone on the neck and provides for moving the roll in and out by means of lugs on the chuck. The bearings used in this case are 10¼ and 9¾ bore by 15¾x10 inches respectively, the bearings at the other end being 10¼x15¾x10 inches. The arrangement gives a neck capacity of 179,000 lb. at 60 R.P.M.

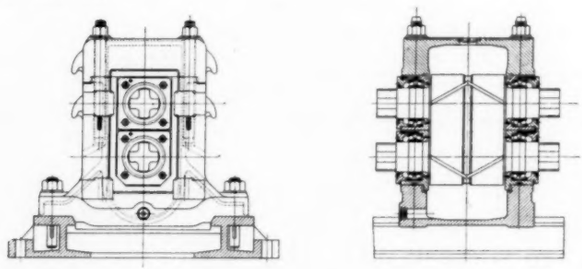


FIG. 7.

The pinion stand mounting for this particular mill is shown in Figure 7. Generally speaking, the application is very simple. The bearings used are 10¼x15¾x4-11/16 inches, with a capacity of 89,000

lb. per neck at 60 R.P.M., or about half that of the roll neck bearings. The bearing chucks are fixed in the housings and the cones given a loose fit on the necks, with a clearance between the cone and fillet ring to take care of pinion float, or any expansion that may occur.

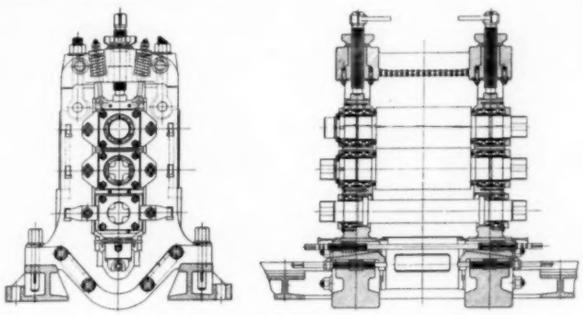


FIG. 8.

Figure 8 shows the general bearing mounting adopted on the five 3 high 16-inch stands in the continuous mill. The mounting on the two working rolls is identical with that on the 18-inch stand, the bearings used being 8½ and 9-inch bore x 14x10 on one end, and 9x14x10 inches on the other. This gives a neck capacity of 145,000 lb. at 50 R.P.M. The mounting for the third, or dummy roll, is very simple, being similar to that used on pinion stands, but using a smaller bearing. The dummy roll is held from end movement by standard clamps. The middle roll is fixed, while the top roll has special lugs on the chucks to facilitate changes in pass alignment and vertical roll adjustment.

At this point it might be interesting to describe the results of a pressure test we made on #7 stand of this mill. This is the fourth of the five 16-inch stands just described. The hydraulic cylinder was used while the mill was rolling #4220 nickel molybdenum steel of approximately .10 to .20 carbon, 1.5 to 2.0 nickel, and .20 to .30 molybdenum. The bar left #6 stand 2¾"x3", was turned 45° for entering #7, which it left as an oval 2-9/32"x4-5/16", a reduction of 22.4% having been effected. The distance between the center lines of the mill screws and bearings is 50¼ inches, and, in order to get the worst possible load conditions the bar was rolled in a pass located 11-13/16 inches from the center line of the bearing under the hydraulic cylinder. A load of 128,000 lb. at 80 R.P.M. was recorded. The bearing capacity in this instance is 136,000 at 80 R.P.M. The bearings in the other stands are continually operating under similar conditions of loading and in the year or more that the mill has been in operation have given no trouble.

Figure 9 shows the mounting details of the pinion stands used with the five 3 high 16-inch mills. The mounting principle is exactly the same as that employed in the 2 high pinion stand for the 18-inch mill. The bearings used are 9x14x4¾ inches with neck capacities of 72,500 lb. at 50 R.P.M.

In addition to these stands, the mill includes a 10-inch finishing stand, which is interesting because the rolls are operated vertically. This mill is designed for very close pass alignment, having a vernier adjustment that gives points of .00025 inches. Wire has been rolled in this mill on a production basis

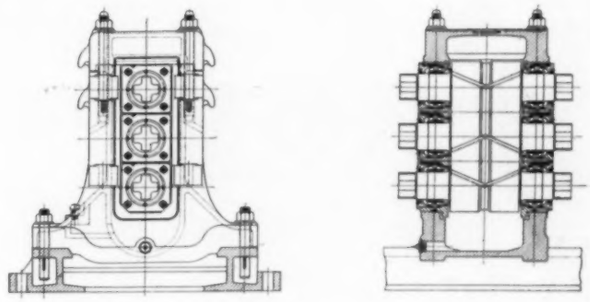


FIG. 9.

that came within tolerances of .001" to .0015", on 5/16-inch round.

Another interesting application, and one that has shown very gratifying results in service, is the bearing installation on the main rolls of our Mannesman seamless tube piercing mill. Before the bearings were installed a pressure test was made to determine the loads encountered while piercing 6-inch rounds to 6x4½-inch tubes. On the driving end the loads were found to be 81,000 lb. per neck, and 72,000 lb. were registered on the opposite end. As a result bearings of 7x12x9-3/16 inches, with a capacity of approximately 103,500 lb. per neck at 90 R.P.M. were installed. This was in September, 1927. In the elapsed period, several sets of rolls have been worn out, but the bearings are still in good condition. As a result of this record, and also because the product showed an improvement in concentricity, it was decided to equip two 36-inch Mannesman mills for our new plant with Timkens.

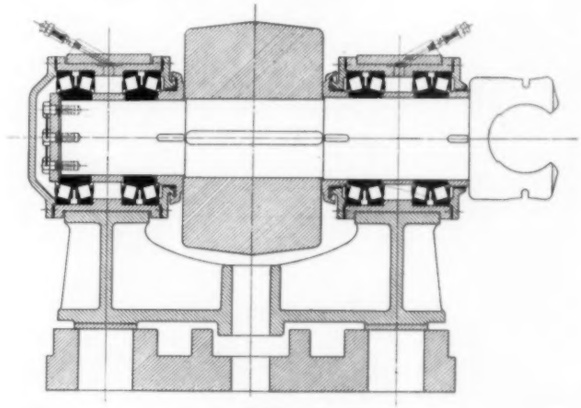


FIG. 10.

Figure 10 shows the method of mounting the bearings on the main rolls of these mills. Bearings of 12x19½x5-7/16 inches are used in groups of two on the driving end, while the same bearings, but with 11-inch bores, are used on the other end. These bearings have a radial capacity of 300,000 lb. per neck at 300 R.P.M. Because of the high speed the bearings are keyed as well as being clamped on the necks. In developing the design, provision was made for ease of roll interchanging, and the closures were equipped with flingers to help keep water and scale out of the bearing chamber. It is not necessary to remove the bearings from the necks when a roll is taken out for dressing, the whole thing coming out

as a unit, as shown in Figure 11. This arrangement is productive of considerable saving in time when rolls must be dressed.

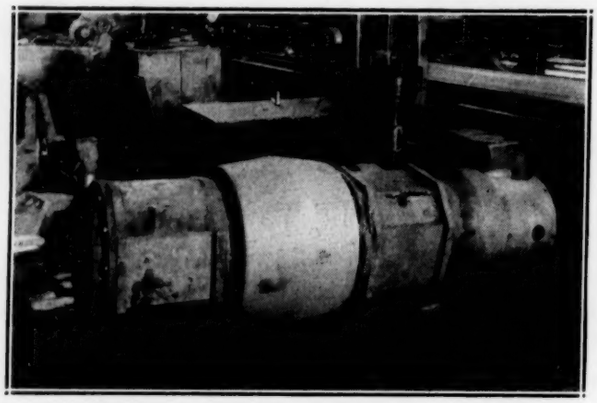


FIG. 11.

Figure 12 shows a bearing mounting for a piercing mill thrust block. A number of these blocks have been in operation in the Timken plant for some time at speeds as high as 500 R.P.M. The design uses a radial bearing mounted at the front of the block, which carries all the radial load imposed by the whip of the plug rod. A suitable thrust bearing is located at the rear end of the block spindle, which, in this particular case is 7x16-7/8x4-3/16 inches with a thrust capacity of 149,000 lb. at 500 R.P.M. The bearings are properly lubricated by maintaining an oil level in the block.

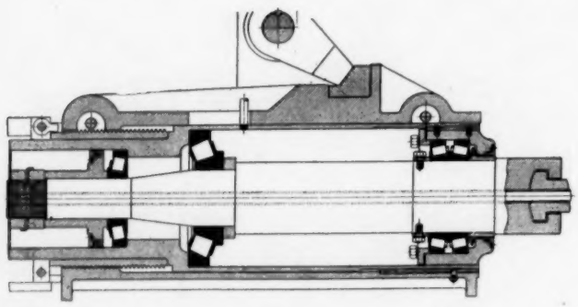


FIG. 12.

The reelers on these mills are also Timken-equipped, as shown in Figure 13. Here also, the cones are keyed to the necks, the mounting in general being the same as that on the piercing mill rolls, except that, since space limitations are not so strict, the mountings can be made self-aligning. Standard 9-7/8x17x11¾-inch bearings are used at each end of the roll, which give a capacity of 175,000 lb. per neck at 300 R.P.M.

Another interesting application is that made to the pinion stand and main rolls of the 2 high, 26-inch re-rolling mill used with the aforementioned piercing mill and reelers. See Figure 14. The pinion stand mounting is practically the same as that used on the stands already described. The bearings in the mill proper are 14x20¼x10¼ inches with a capacity of 181,000 lb. per neck at 85 R.P.M. Here also provision has been made for pass alignment, using the principles of design already discussed in the case of the 22-inch mills.

In addition to the applications in the Timken steel mill, several interesting applications have been made in other plants, for different purposes. One of these, which has given very gratifying results, is that made to the roll and pinion necks of a Belgian type mill rolling copper wire. The installation comprises a 3 high pinion stand driving one stand of 3 high, and one of 2 high 20-inch mills that make up the roughing train, and two pinion stands driving 9 stands of 14 and 12-inch intermediate and finishing mills. The bearings in the roughing mill are similar to those used in the 22-inch Timken mill already described, that is 12x17½x9½ inches on the 3 high, and 12x17½x4⅜ inches on the 2 high mill. On the 3 high pinion stand two single bearings 12x17½x2½ inches per neck are mounted in a way similar to that employed on the pinion stand of our 28-inch mill. All the other mills have two 6x10x2⅝-inch bearings per neck with a resultant neck capacity of 27,000 lbs. at 300 R.P.M. It is of interest that the synchronous motors driving these mills are also equipped with Timken bearings.

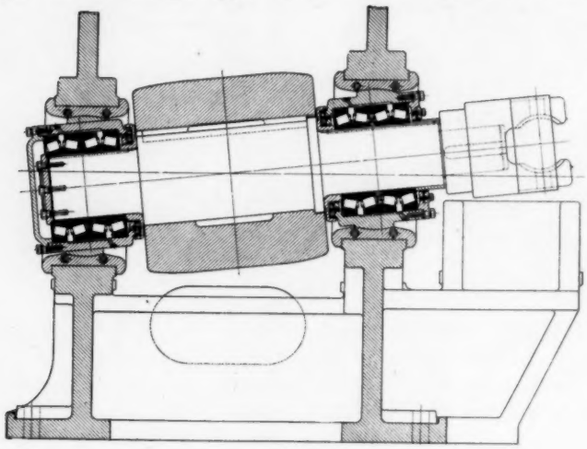


FIG. 13.

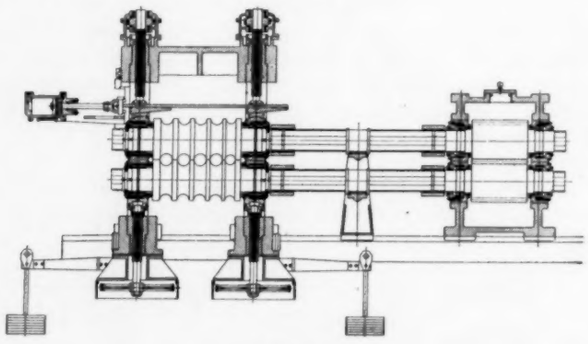


FIG. 14.

Another complete copper mill installation recently put in service consists of one 3 high pinion stand and one 3 high 22-inch breakdown mill using the same bearings as the mill just described. In addition there are 14 stands of 14 and 12-inch intermediate and finishing mills, with their pinion stands, where the bearings are also similar to those in the

copper mill just mentioned, i.e., 6x10x2⅝ inches. The gear drives for this mill are also equipped with the same size bearings.

The application of Timken bearings to the rolls of a vertical edging mill used in connection with a 21-inch skelp mill is also of interest. The method of mounting is shown in Figure 15. A double bearing

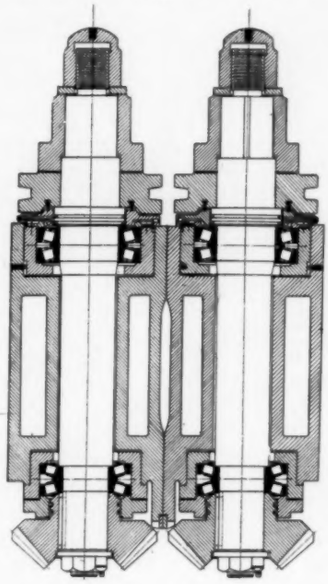


FIG. 15.

9x16¾x7 inches with a capacity of 160,000 lb. at 100 R.P.M. is used adjacent to the edging roll. The bearings are mounted in a cartridge that floats in the main housing. Two bearings 7x14x2-7/16 inches, with a capacity of 95,000 lb. at 100 R.P.M. are mounted at the bottom of the roll spindle in a cartridge bolted to the main housing. These bearings fix the position of the main rolls and bevel gears as well as carrying the thrust loads of the rolls and gears. Piston ring closures are used with these bearings to prevent the escape of the grease. The closure in the roll proper consists of a circular dam and has proved very effective in keeping water and scale out of the upper bearings. In addition, piston rings are also used between the shaft and the plate that encloses the bearing box. The bearings are given a tight fit on the shafts that carry the rolls, since it is unnecessary to remove them when a roll change is made. The bearings are adjusted by means of shims, and all bearings are lubricated through pressure fittings.

Gear Drives

Under present conditions of steel mill practice, the application of anti-friction bearings to gear drives is almost as important in its general effect as it is in the case of roll necks. This statement is borne out by the fact that in many cases where the mills themselves run on plain bearings for one reason or another, the gear drives are equipped with Timkens.

In the Timken mill a 1500 H.P., double flywheel, single reduction herringbone unit is being used to drive the 28-inch mill. The flywheels weigh 26,000 lbs. each. The pinion shaft of this unit, which is shown in Figure 16 has a speed of 360 R.P.M., and

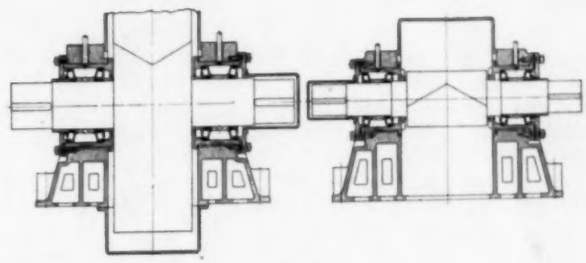


FIG. 16.

the main shaft a speed of 50 R.P.M. Two bearings 12x19½x3 inches are used on each end of the pinion shaft, and two bearings 17x26½x3½ inches on each end of the gear shaft. The general design is perfectly apparent from the cut.

The 1000 H.P. unit used to drive the 2 high 26-inch re-rolling mill is of exactly the same design. Here the pinion shaft carries two 15,000 lb. flywheels at 250 R.P.M., and has two 10¼x15¾x2¾-inch bearings per position. The gear shaft has two 13½x21x3-inch bearings at each end.

In making bearing load calculations for gear drives 200% motor overload, that is, 300% motor rating is always used as a basis for computations. In the case of flywheel units the horsepower developed by the energy given up by the flywheel is also included. Where data is available the energy given up by the motor rotor is also taken into account. The latter will, of course, vary with the type of motor, and the percent slip. As a concrete example, there is the case of a large unit for which our bearings were recently supplied. The motor is rated 1600 H.P., and is designed to pull 3400 H.P. with 15% slip. The flywheels, which weigh 27,000 lbs. each, can deliver 4200 H.P. during a 15% slip. The motor rotor develops 500 H.P. The unit was consequently figured for 3400+4200+500, or a total of 8,100 H.P. As a result, two bearings per box 16x26½x3½ inches were used on the pinion shaft, 17x26½x3½ inches on the intermediate, and 21x30-7/8x3½ on the take-off shaft. This gives capacities of 286,000 lbs. at 360 R.P.M., 377,000 lbs. at

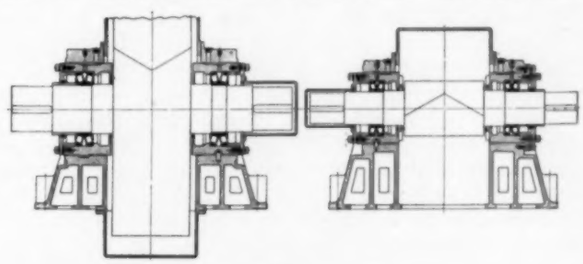


FIG. 17.

146 R.P.M., and 540,000 at 30 R.P.M. for the respective shafts. The bearings are very conservatively loaded. The intermediate boxes are the most heavily loaded, but even here, when the unit is developing its maximum, 8100 H.P., the bearings are only loaded 84%. This, incidentally, is the first large, double reduction unit to be equipped with Timken bearings.

Figure 17 shows the single reduction 1500 H.P., 360/70 R.P.M. unit which serves the 2000 H.P. motor

used to drive our 22-inch mill. The same size bearings are used as those used in the 28-inch mill unit, but the design is somewhat different. In this case double cone bearings are used, instead of single cone bearings separated by a spacing ring. Since this unit has no flywheels, the bearings are not nearly as heavily loaded.

In addition to the above there is also in our own mill a large number of units used to drive the 16 stand continuous mill. These are all of the single helical gear type in ratings varying from 500 to 1200 H.P. Some are single and some are twin drive type. A typical bearing mounting is shown in Figure 18.

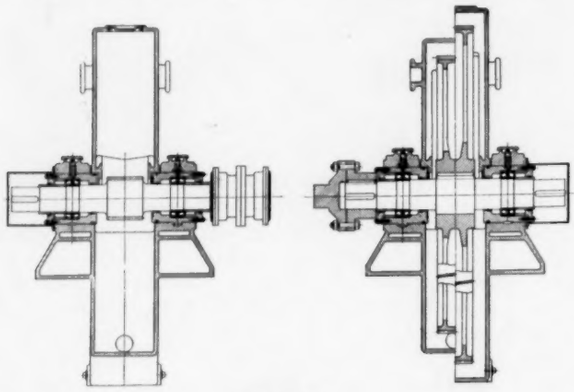


FIG. 18.

In this case one bearing box is positively fixed against end movement, while the other is allowed to float to take care of inaccuracies in machining, or expansion. The original design of these units called for individual lubrication of gears and bearings. A short period of operation showed, however, that they worked best, and required the least attention when the bearings were splash lubricated from the gears. Metal baffle plates were installed to catch the splash from the gears, and pipe connections were installed to carry the accumulated oil to the bearings. The bearing boxes have been equipped with overflows that maintain a constant level of oil around the bearings, and drain the excess back into the case.

This system has been installed on all of our gear drives, except that for the 28-inch mill. In this particular instance, a small pump, driven by the low speed shaft, circulates oil to all bearings from a separate source of supply. It is essential that an oil level be maintained continuously in bearing boxes so that they will not dry out during the time that the unit is coming up to speed and the pump has not been able to establish circulation.

It is recommended on all installations using this system that the bearing boxes be provided with sufficient storage capacity to maintain a proper oil level at all times. Aside from this feature the general mounting recommendations are the same as those shown in the three preceding figures.

Another interesting gear drive application is a triple reduction unit used to drive one of the billet shears in the Timken plant. See Figure 19. The bearings are all mounted in cartridges, and on each shaft one cartridge is fixed to the gear case, while the other is permitted to float to take care of expansion, or inaccuracies in machining. Since the

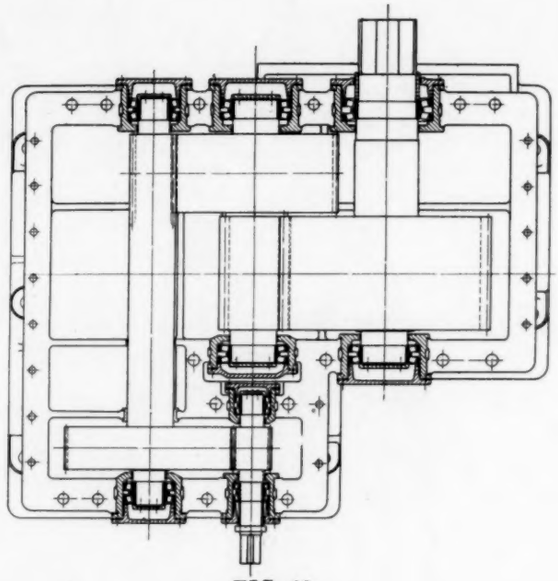


FIG. 19.

low shaft speeds do not permit splash lubrication from the gears, all bearing boxes are completely enclosed, and bearings are independently lubricated with grease. The bearings are adjusted in the cartridges before the gears are mounted in the case, thus simplifying assembly considerably. The cartridges are fitted into the case .002 to .004 loose.

The figures that have been shown are typical of the gear drive mountings used by the leading gear drive manufacturers. In all, close to 100 Timken equipped mill drives varying in motor rating from 100 to 2000 H.P. have been built, and are operating very satisfactorily. Figure 20 shows a typical method of mounting Timken Bearings on rolls of rolling mills.

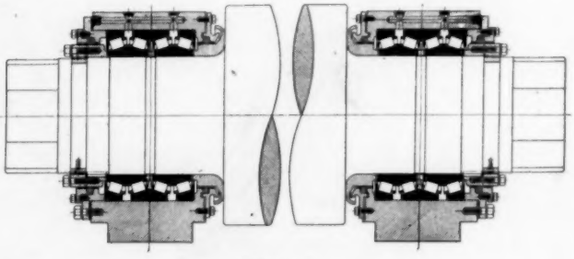


FIG. 20.

The synopsis given in the notice of this meeting stated that information will be given concerning the power savings effected through the use of our bearings.

It was the hope that this could be given from tests which were conducted on our new anti-friction mills at Canton, but due to conditions which were beyond all control, these have not yet been run.

The speaker, however, anticipated this possibility, and rather than disappoint you wishes to present an analysis he made of a report by Dr. Puppe on the later tests he ran on some German mills for the purpose of determining rolling pressures at the same time power consumptions were observed.

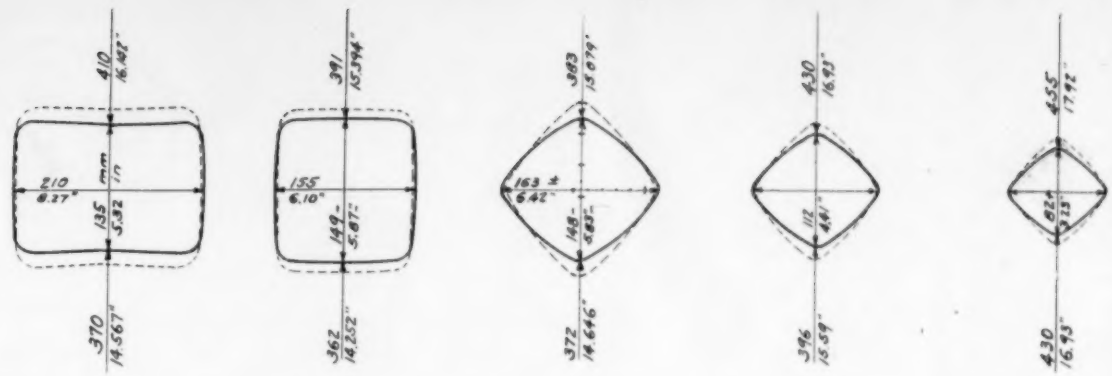
TEST NUMBER — 1

570 mm (22.44") 3 HIGH MILL

ROLLING :— 50mm² (1.97" × 1.97") BILLETS
FROM 406.5 kg (896.2#) BILLET.

Neck dia 290 mm (11.42").
Area of neck bearing 44.954 in² 290 cm²
Lgth " " " 300mm (11.81")

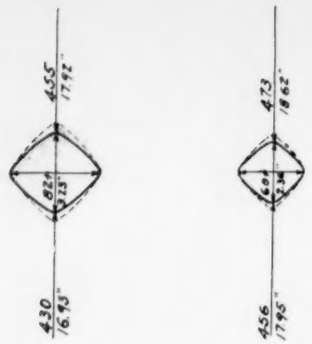
"Roughing Stand"



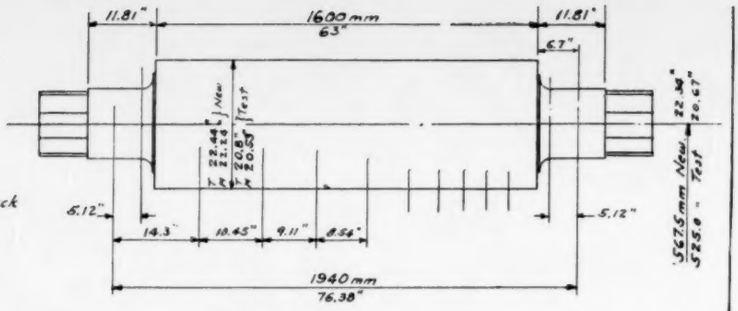
H = 168mm 6.614" | 177mm 6.97" | 182mm 7.166" | 143mm 5.63"

1	Pass Number		2		4		6		8		10	
			Metric	English	Metric	English	Metric	English	Metric	English	Metric	English
2	Time of Pass in Seconds "t"		.82		.90		1.19		1.80		3.10	
3	Speed range in R.P.M.		120.3-118.1		116.2-114.8		114.7-111.9		112.8-109.0		112.3-107.8	
4	Average Speed in R.P.M.		119.1		115.5		113.0		110.6		110.0	
6a	Area of preceding pass	mm²	35000	54.25	25200	39.06	20000	31.00	13300	20.62	8000	12.40
6	" " this "	mm²	29520	45.76	23080	35.80	17520	27.16	11180	17.33	6340	9.83
7a	Reduction in Area, preceding pass	mm²	—	—	4350	6.73	3040	4.71	4220	6.54	3170	4.91
7	" " " this pass	mm²	5450	8.45	2090	3.24	2520	3.91	2120	3.28	1670	2.59
8a	Length of billet before this pass	m	1.481	4.86	2.057	6.75	2.584	8.48	3.893	12.77	6.465	21.21
8	" " " after " "	m	1.754	5.75	2.243	7.36	2.955	9.70	4.632	15.20	8.168	30.08
9	Volume displaced during this pass	mm³	807500	492.36	4299100	262.3	6511700	397.2	8253200	503.45	10796600	658.59
10	Average pressure on outer roll neck	kg	20575	45355	24814	54700	44644	98413	64641	142600	66616	146850
11	" " " inner " "	kg	95253	209975	55147	121565	58376	128693	53997	119030	37341	82314
12	Maximum " " outer " "	kg	23060	50834	28293	62368	50212	110686	71375	157340	74222	163614
13	" " " inner " "	kg	113680	250596	73356	161705	67895	149000	60680	133760	42212	93050
14	Total Ave. pressure on both necks	kg	115828	255330	79961	176265	103020	227100	118690	261630	103960	229164
15	" Max " " " "	kg	136740	301430	101650	224073	117810	259700	132060	291110	116430	256640
16	Volume displaced in mm³ / k × Total Ave. Press. in kg (Line 9 / Line 2 × 14)		85.1	.00736	59.7	.00165	53.1	.00147	38.7	.00107	33.5	.00092
17	Temperature of steel before pass	°C.	1203	2197.4	1189	2172.2	1162	2123.6	1156	2112.8	1150	2102
18	Energy furnished to the Motor	HP	286	282	373	367.9	477	470.5	534	526.7	575	567.1
19	Average Motor output	HP	259	256	343	338.3	443	436.9	496	489.2	535	527.7
20	Maximum " "	HP	355	350	405	398.5	513	506.0	611	602.6	588	580.0
21	Energy from Motor incl. light load losses	H.P.sec.	212	209.1	309	304.8	527	519.8	894	881.8	1657	1634.3
22	Light load losses during the pass	H.P.sec.	162	159.8	177	174.6	234	230.8	354	349.2	610	601.6
23	Energy from Motor excl. light load losses	H.P.sec.	50	49.3	132	130.2	293	289.0	540	532.6	1047	1032.7
24	Ave. energy given up by the rotating masses	HP	1730	1706.3	862 (873.3)	850.2	1526 (1642) / 1716	1505	1225	1208.2	862	850.2
25	Energy from rotating masses during pass	H.P.sec.	1418	1398.6	775 (876)	764.4	1816	1791	2208	2177.8	2667	2630.5
26	Ave. energy used incl. light load losses	HP	1989	1961.8	1205 (1362)	1188.5	1969 (1838) / 1839.4	1942	1721	1697.4	1397	1377.9
27	" " " excl. " " "	HP	1792	1767.5	1008 (1139.4)	994.2	1772	1747.7	1524	1403.1	1200	1183.6
28	Energy during pass incl. L.L. Losses	H.P.sec.	1630	1607.7	1084 (1183.5)	1069.1	2343 (2343)	2311.9	3102	3059.5	4324	4264.8
29	" " " excl. " "	H.P.sec.	1468	1447.9	907 (1006.5)	894.6	2109 (2049)	2080.1	2748	2710.4	3714	3663.1
30	% of Total Rolling Work done by rotating masses		96.5		85.5		86.2		80.5		71.8	
31	Volume displaced in mm³ / Total Rolling Work in mkg (Line 29 × 75)		73.3	.3401	63.1	.293	41.2	.191	40.0	.186	38.7	.1796
				2.94		3.41		5.23		5.34		5.55
32	Bending Mom in Mid'l roll at ctr pass	Mm mt	39.70	344,853	43.10	374,080	86.4	489,520	63.2	548,760	51.7	448,760
33	" " " Top " " "	Mt mt	38.95	338,745	42.00	364,600	85.2	479,360	61.95	537,260	50.5	438,400
34	Total Twisting Moment	Mt mt	10.76	93,940	6.25	54,500	11.23	97,764	9.86	85,850	7.82	67,776
35	Tw. Mom in Mid'l and Top Rolls .5 Mt	mt	5.38	46,970	3.125	27,250	5.61	48,382	4.93	42,925	3.91	33,888
36	Combined Mom. Mid'l Roll ctr pass	Mc mt	39.85	345,782	43.68	379,400	86.45	489,860	63.91	556,390	51.7	448,560
37	" " Top " "	Mc mt	39.07	339,280	42.00	364,600	85.22	479,096	61.90	537,920	50.58	439,344
38	Section Modulus ctr pass Mid'l Roll	W cm³	4973	303.35	4657	284.08	5054	308.3	6096	371.9	7806	476.2
39	" " " " Top Roll	W cm³	6766	412.73	5868	357.95	5515	336.4	7806	476.2	9247	564.1
40	Fibre stress in Mid'l Roll	kg/cm²	802	11400	938	13342	1117	15887	1095	15574	663	9430
41	" " " Top "	kg/cm²	578	8214	716	10184	1001	14237	795	11307	547	7780
42	Specific bearing pressure outer necks	kg/cm²	70.9	1008.4	85.6	1217.5	153.9	2189	223	3172	229.9	3270
43	" " " inner "	kg/cm²	328.5	4672.3	190.1	2703.8	201.1	2860	186.1	2647	128.7	1830.5
44	Rubbing velocity "V" of roll necks	m	1.81	5.94	1.784	5.75	1.751	5.74	1.678	5.505	1.469	5.476
45	Outer neck pressure k × v	kv in the text	128.2	5990	150.1	7004	263.9	12315	374.0	17452	383.5	17896
46	Inner " " k × v	kv " " "	595.0	27760	332.0	15500	345.0	16099	312.0	14560	214.5	10010
47	Roll Neck Friction loss	HP	418	412.3	280.4	276.6	353.2	347	396.0	390.6	346.8	342
48	Energy used for Roll Neck Friction	H.P.sec.	342.8	338.1	252.4	248.9	426.0	414.2	716.4	706.6	1076.0	1061.3
49	" " " Pinion Stand Losses	H.P.sec.	88.08	86.87	54.47	53.67	126.5	124.6	164.9	162.6	232.8	229.7
50	Net Rolling Work Eu	H.P.sec.	1037.1	1022.9	600.2 (699.9)	592.0 (690.3)	1562.5 (1443.8)	1541.1	1866.7	1841.1	2415.0 (2381.9)	2381.9
51	Volume displaced in mm³ / Net Rolling Work in mkg (Line 50 × 75)		103.8	.4816	95.6 (81.0)	.4436 (.38)	55.6 (59.2)	.258 (.275)	59.0	.2738	59.6	.2765
				2.08		2.24 (2.43)		3.88 (3.64)		3.65		3.61
52	% of Total Energy input used by motor itself		9.50		8.00		7.30		7.10		7.00	
53	% " Energy used during pass for Light load losses		9.95		16.32		9.98		11.41		14.11	
54	% " " " " " by Pinions		5.40		5.01		5.40		5.29		5.16	
55	% " " " " " for Neck Friction		21.00		23.20		17.90		23.00		24.80	
56	% " " " " " Actual Displacement		63.70		55.40		66.72		60.20		55.90	

TABLE NO. 1.



Section Modulus of neck
.0982 d^3 = 146.25



Cast Steel Rolls.
Allowable Fibre Stress 750 kg/cm² (10668 #/")

$\frac{250596 \times 5.12}{146.25} = 8800$ #"

$\frac{250596 \times 6.7}{146.25} = 11500$ #"

10		12		The Metric value multiplied by a conversion factor gives the English value in each case.			1
Metric	English	Metric	English.	Conv. factor	English value in terms of:-		
3.10		5.48					2
112.3 - 107.8		110.9 - 107.4					3
110.0		109.3					4
8000	12.40	4530	7.02	.00155	Square inches.		6a
6340	9.83	3700	5.74	"	" "		6
3170	4.91	1810	2.81	"	" "		7a
1670	2.59	830	1.29	"	" "		7
6.465	21.21	11.431	37.50	3.2809	Feet.		8a
8.168	30.08	13.995	45.92	"	"		8
10796600	658.59	9487800	5805 -	.000061	Cubic inches.		9
66616	146850	52732	116242	2.2046	Pounds.		10
37341	82314	21818	48096	"	"		11
74222	163614	56212	123913	"	"		12
42212	93050	22654	52143	"	"		13
103960	229164	74550	164338	"	"		14
116430	256640	79866	176056	"	"		15
33.5	.00092	23.4	.00065	.0000277	Cu.in per pound pressure per second.		16
1150	2102	1150	2102	°/5 C + 32	°Fahrenheit.		17
575	567.1	543	535.6	.9863	Horsepower		18
535	527.7	504	497.1	"	"		19
588	580.0	561	553.3	"	"		20
1657	1634.3	2762	2724.2	"	" Seconds.		21
610	601.6	1080	1065.2	"	" "		22
1047	1032.7	1682	1659.0	"	" "	21 - 22	23
862	850.2	377	371.8	"	"		24
2667	2630.5	2066	2037.7	"	" "	24 × "t"	25
1397	1377.9	881	868.9	"	" 24+(21+t) = 24+19	Horsepower Rate for entire Unit.	26
1200	11836	694	674.6	"	" 26-(22+t)	" " " Total Rolling Work.	27
4324	4264.8	4328	4761.9	"	" Seconds. 26 × t	Total Energy used during Pass.	28
3714	3663.1	3748	3696.7	"	" " 27×t" 25+23 28-22	Total Rolling Work "E"	29
71.8		55.2					30
38.7	.1796	348	.1615	.00464	Cu.in per #sec. Total Rolling Work "E"	Reciprocal = #sec. per Cu.in displaced.	31
	5.55		6.18				
51.7	4,487,560	31.7	2,752,560	86797.3	Inch pounds		32
50.5	4,383,400	30.7	2,664,760	"	" "		33
7.82	678,776	4.47	387,996	"	" "		34
3.91	339,388	2.235	193,998	"	" "		35
51.7	4,487,560	31.7	2,752,560	"	" "		36
50.58	4,390,344	30.75	2,669,100	"	" "		37
7806	476.2	9308	567.8	.061	Inches³		38
9247	564.1	10386	633.5	"	"		39
663	9430	341	4850	14.2232	Pounds per sq. in.		40
547	7780	296	4110	"	" " "		41
2299	3270	1818	2586	"	" " "		42
1287	1830.5	755	1072	"	" " "		43
1.669	5.476	1.658	5.44	3.2809	Feet per Second.		44
383.5	17896	380.8	14037	46.665	Feet pounds per Second	• (kg/cm² × 14.2232) = (m × 3.2809)	45
214.5	10010	1249	5928	"	" " " "		46
946.8	342	247.4	249.0	.9863	Horse power		47
1076.0	1061.3	1356.0	1337.4	"	" Seconds 47 × t		48
222.8	219.7	229.9	231.8	"	" "		49
2415.0	2381.9	2167.0	2137.3	"	" " 29-(48+49)		50
59.6	.2765	58.4	.271	.00464	Cu.in per #Sec. Net Rolling Work "Eu"	Reciprocal = #Sec. per Cu.in. displaced.	51
	3.61		3.69				
7.00		7.10				(18 - 19) ÷ 18	52
14.11		23.40				22 ÷ 28	53
5.16		4.67				49 ÷ 28	54
24.80		24.10				48 ÷ 28	55
55.96		45.80				50 ÷ 28	56

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The purpose in making this analysis was that of learning, if possible, the accuracy of Dr. Puppe's calculated neck friction losses, and from these to approximate the savings that could be effected if roller bearings were used in one of the German mills.

The final object was of course that of comparing the approximated savings with the actual saving disclosed by the tests we ran on our own 16" three high roughing stand at Canton, which saving, you will recall, ranged from 24.5% to 51%, the average being about 36%.*

The particular tests by Dr. Puppe to be discussed tonight were made in 1910. They covered the rolling of billets, angles, channels and I Beams, and should not be confused with the tests that were conducted in 1906 and reported in complete detail in Dr. Puppe's book "Power Required for Driving Rolling Mills," a copy of which you no doubt all possess.

The report of the 1910 tests was published in part in the January, 1914, issues of Stahl und Eisen. Numerous charts and diagrams were shown as was also one table on billet rolling. This table gave an abundance of data, but all values were expressed in metric terms. (See Table No. 1).

Since we are accustomed to using English measurements entirely, and find it more or less confusing to deal with the metric, the speaker has converted these values into English. The table, however, shows the values in both terms, as well as the conversion factors he used when the English values were calculated.

The extreme thoroughness on the part of Dr. Puppe and his associates and the sequence in which the data was compiled, must at once appeal to you as representing a tremendous amount of work. You will surely agree that, considering the facilities they had at their disposal at the time, they have made a valuable contribution to the general knowledge that exists of the art of rolling steel.

The tests were conducted in Peine, Germany, on four sizes of three high mills. Three of these, however, were steam engine driven but, due to the difficulty of accurately recording the beginning and ending of the passes, only rolling pressures were taken in the mills so driven.

The fourth mill was motor-driven which permitted the taking of power readings as well as rolling pressures.

Eighteen tests, in all, were made. These covered the rolling of 101 billets and blooms, of which number, forty-six were rolled in a 570 mm (22.44") three high motor-driven unit.

The drive for this unit was a 500-volt D.C. motor with a speed range of 90 to 120 R.P.M. The output of the motor for this range of speed was 900 H.P. continuous, 1200 H.P. for three minutes and 1650 H.P. momentary peaks.

The flywheel for this drive was 7500 mm (295.3") in diameter; it weighed 43180 Kg. (95190 lbs.) and had a moment of inertia, including that of the other rotating masses, of 37000 Kgm. sec.^{2} (267600 ft.^{2}lbs.)

Extreme care was taken to obtain exact roll speeds during the passes. From these, the energy delivered by the rotating masses during anyone pass was calculated with the formula

$$E \text{ in Kgm} = \frac{J}{2}\left[\left(\frac{pi\ n_1}{30}\right)^2 - \left(\frac{pi\ n_2}{30}\right)^2\right]$$

in which J = the moment of inertia of the rotating masses in terms of the mass, i. e., Mr^2 = 37000 Kgm Sec.^{2}. n_1 and n_2 = the speeds of the rolls at the beginning and ending of the passes in R.P.M.

In order to facilitate the energy calculations all possible, the speaker has reduced this formula to its simplest form, i. e., since one metric horsepower second equals 75 Kgm, the formula can be written

$$E = \frac{\frac{J}{2}\left[\left(\frac{pi\ n_1}{30}\right)^2 - \left(\frac{pi\ n_2}{30}\right)^2\right]}{75} =$$

$$2.705\ (n_1^2 - n_2^2) \text{ in H.P. sec. (Metric)} \quad (1)$$

The tests made on the motor-driven unit were run while as many as eight billets were rolled. In some of the tests on the larger mills, as many as eleven were rolled, but in no case were fewer than four billets used. In no instance was a test made on a finishing pass.

Table No. 1 shows the results obtained while only one billet was rolled. You will at once appreciate that to have shown the results for all billets would have entailed many days of work to make the necessary conversions from metric into English. It is therefore hoped that the data given for one billet will suffice tonight to show the scope of the work covered by the tables in Dr. Puppe's report.

Time will not permit of describing the data entered upon every line. The headings are self explanatory. The heading to the left, at the top, describes the mill, its position in the unit, and the product rolled. The cuts above the columns representing the passes are reproduced from those given in the report. The full lines denote the section delivered by the pass. The dotted lines indicate the relative size and shape of the bar before entering. The latter, of course, are the sections delivered by the preceding (or odd numbered) passes.

Only the passes which occurred between the top and middle rolls are shown. This is due to the fact that the hydraulic cylinders could record the pressures for those passes only. It is regretted that it was not possible to take pressures for the passes between the middle and bottom rolls as well. It would be interesting to follow a billet during its entire rolling operations in order to obtain the pressures and power consumption for every pass made.

It should be noted that the recording devices used by Dr. Puppe enabled him to determine the time of a pass to within one one-hundredth of a second. The ranges in roll speeds are given to the nearest tenth of a revolution per minute. The average speed of the rolls given is not the arithmetic average, but that obtained by planimetric measurements of the speed charts that were taken.

Although both the average and maximum rolling pressures are given for each pass, the average pressure only was used when the horsepower requirements for roll neck friction were calculated. It ap-

*See "Analysis of Power Consumption Test 16" Three High Roughing Stand, Timken Steel Mill." Iron & Steel Engineer, August, 1928.

pears however that the maximum pressures were used when the stresses in the roll bodies were determined.

In order to show the difference between the fibre stress in the roll body and that in the necks, the speaker has shown calculations for neck stresses on the table. The two distances given from the point of application to the point of greatest bending are shown only for the purpose of comparison since there exists a difference of opinion as to which lever arm should be used. The longer arm shown is that actually obtained from the roll designs. The shorter arm is given as closely as could be approximated, no radius of the neck fillet having been given on the roll drawings.

Line 34 gives the total twisting moment in the stand that was rolling, that is, the values represent the torque in the two working rolls for everything but the live load losses as can be checked with the formula: $\frac{T\,N}{63025} = H.P.$

Referring to Pass 2, Test 1, $\frac{933940 \times 119.1}{63025} = 1767$ H.P. which checks with line 27 which gives 1767.5 H.P. for the average energy used excluding that for live load losses.

While it is true that the pinion stand losses could also have been omitted when the torque in the working rolls was calculated, the correction would be very small. As it is, the twisting moment per roll is so low, as compared with the bending moment, as to show very little increase in the combined moment over the bending moment alone. This fact, no doubt, indicated that it was not thought necessary to calculate the torsional stresses in the roll bodies.

Time will not permit of making a complete review of the data that appears on every line. We can however devote a few moments to the power consumption for a pass, and its distribution over the four items: light load, pinion stand and neck friction losses, and the actual work of displacement.

The total power used per pass, as given in line 28, is of course that furnished by the motor and the rotating masses which appear in lines 21 and 25.

The distribution of this power is given in lines 53 to 56 inclusive, all percentages being given in terms of the actual energy supplied by the motor and the rotating masses.

You will note that line 52 gives the percent of the total energy input used by the motor itself, and that only the motor output is charged to the mill.

The light load losses (line 22), it can be shown, are always quite the same. They check out at around 194 H.P. for each pass.

The pinion stand losses (line 49) were fixed at between 5% and 6% of the total energy used for a pass. This figure was established by Dr. Puppe from data obtained from tests he made to determine pinion stand losses, a complete report of which appeared in the April and May, 1911, issues of Stahl und Eisen.

The roll neck friction losses (line 48) are values over which more or less discussion can be held. They are calculated on a basis of .075 for f, the coefficient of friction, and which with the formula $\frac{K\,n\,pi\,r\,f}{30}$ gives the neck friction loss for one roll in cm/kg/per second.

K—represents the total average rolling pressure in kg.
n —represents the average roll speed in R.P.M.
r —represents the radius of the roll neck in cm.
f —represents the coefficient of friction .075 (taken after Morin).

From the above it follows that the neck friction loss rate for the working rolls must be:

$\frac{2\,K\,n\,pi\,r\,f}{30 \times 7500}$ in Horsepower (Metric)

The 7500 being the number of cm/kg/per second in one horsepower.

The calculations for the horsepower rate for roll neck friction can be expedited considerably by the use of the rubbing velocity given in line 44, it being only necessary to use the formula $\frac{2\,K\,v}{1000}$. (2)

This formula is merely a simplification of the neck friction horsepower rate formula (with the neck diameter taken in mm).

$2\ K. \frac{pi\ d}{1000} \cdot \frac{n}{60} \cdot \frac{.075}{75}$ in which the quantity $\frac{pi\ d\ n}{1000 \times 60}$ represents v, the rubbing velocity in meters per second, the division $\frac{.075}{75}$ yields the divisor 1000, and the formula becomes $\frac{2\,K\,v}{1000}$ in horsepower (Metric) for neck friction.

When calculating the horsepower (English) for roll neck friction the formula $\frac{2\,K\,f\,v_1}{550} = H.P.$ (English) (3) should be used.

In the above formulae (2) and (3)

K = Average rolling pressure $\begin{cases} \text{in Kg for formula (2)} \\ \text{in lbs. for formula (3)} \end{cases}$

f = Coefficient of friction .075
v = Rubbing velocity in meters per second in (2)
v_1 = Rubbing velocity in feet per second in (3)

Line 50 finally lists the energy used for the actual displacement of the steel. This is merely the remainder of the total energy used during a pass (line 28) after the other losses are deducted.

While working on this analysis, the speaker, upon arriving at this point, pondered over the thought that if the horsepower Dr. Puppe gives for the actual displacement could now be checked by calculation it would add considerably to the satisfaction so far derived from the hours of work he had put on this report. He however did not know of any generally accepted formula that is used by the trade and attempted to apply such data as has come to his knowledge through articles which appeared in trade journals.

Two rather reconcilable theories were found to have been published. The first was advanced by Mr. G. A. Merkt who gives deductions, based upon tests, in an article he published in the Iron Trade

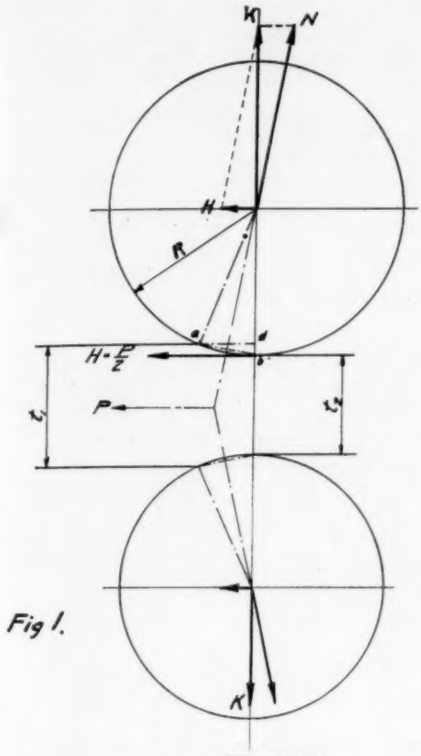


Fig 1.

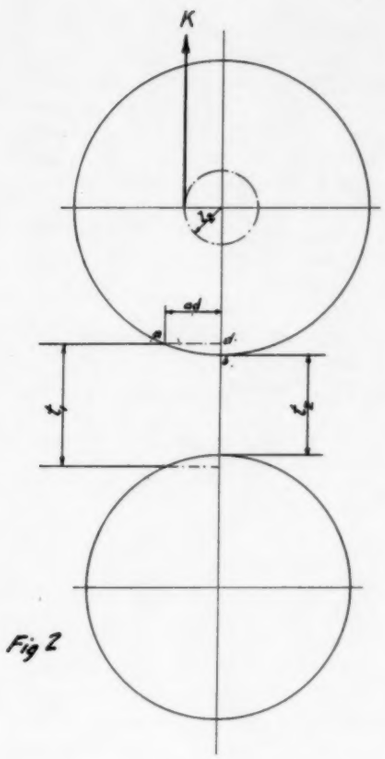


Fig 2

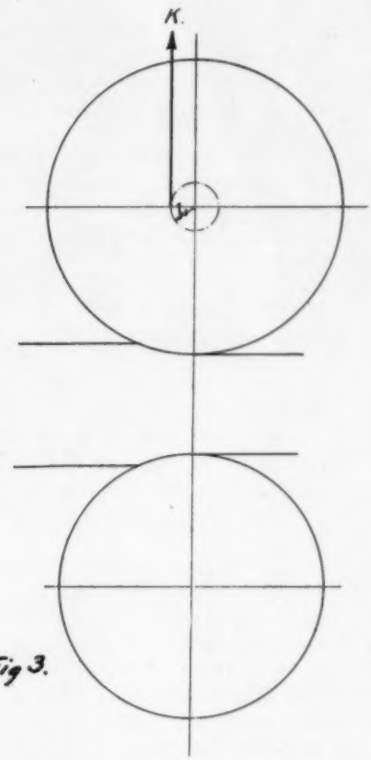


Fig 3.

$K = \sqrt{N^2 - H^2}$

$H = \frac{P}{2}$ $\qquad N = P\sqrt{\frac{R}{t_1 - t_2}}$

$K = \sqrt{\left(P\sqrt{\frac{R}{t_1 - t_2}}\right)^2 - \left(\frac{P}{2}\right)^2}$

$K^2 = P^2\frac{R}{t_1 - t_2} - \frac{P^2}{4}$

$P = \sqrt{\frac{K^2}{\frac{R}{t_1 - t_2} - .25}}$

Tw. Mom. for one roll $= R \times \frac{1}{2}\sqrt{\frac{K^2}{\frac{R}{t_1 - t_2} - .25}}$

$l_2 = .7ad$

Tw. Mom. for one roll $= K \times .7ad$

$255330^{\#} \times (.7 \times 3.08") = 550490^{"\#}$

$8.071 \times .5\sqrt{\frac{255330^2}{\frac{8.071}{1.2216} - 25}} = 408680^{"\#}$

$50635^{\#}$ for H

$\frac{1022.9 \times 63025}{82 \times 2 \times 1191} = 329375^{"\#}$

$\frac{329375}{255330} = 1.29" = l_1$

$\frac{329375}{8.071} = 40810^{\#}$ for H

PLATE I

F. Waldorf, Pittsburgh 12-16-29

Review of April 22, 1909, under the heading: "Analysis of Power Consumption in a Roll Pass." The second appeared in articles entitled "Rolling Mill Calculations" by Mr. J. D. Kellar which were published in the Iron & Steel World in 1927, although it appears that they referred primarily to blooming mills.

The slide showing Mr. Merkt's analysis is a reproduction made from a photostat the Iron Trade Review very kindly furnished. Many of you will recall reading it, but because of several minor errors in the way the compositors (who are not necessarily supposed to be engineers) set up some of the formula,** found it difficult to follow Mr. Merkt's line of reasoning and put off a thorough perusual of the analysis until a future day which never arrived.

Figure 1, Plate 1, shows the distribution of the loads according to Merkt, as well as the formula the author has used to determine the force H at the circumference of the roll from Puppe's rolling pressure (K) since the value of N is unknown.**

**Due to the length of Mr. Merkt's article, it cannot be reproduced at this time. The reader should refer to the original text and use the following construction of the several formulae:

The quantity $\sqrt{P(t_1 - t_2)}$ in equation 3 should have been shown $\sqrt{R(t_1 - t_2)}$; and the final formula $N = \sqrt{\frac{R}{t_1 - t_2}}$ should read $N = P\sqrt{\frac{R}{t_1 - t_2}}$

In his Figure 4, the quantity $\left(R - \frac{t_1 - t_2}{2}\right)$ in the formula $ad = \sqrt{R^2 - \left(R - \frac{t_1 - t_2}{2}\right)}$ should have been shown squared i.e. $\left(R - \frac{t_1 - t_2}{2}\right)^2$

If the formulae in Mr. Merkt's analysis are used as corrected** little trouble will be experienced in applying it to Dr. Puppe's first two passes. These are the only ones giving dimensions enough to permit of making a check on the horsepower rate for displacement.

You will note that Merkt uses a force P when calculating this horsepower, P being the horizontal component of the two normal pressures N on the steel in the rolls. This normal is shown as bisecting the arc ab. Dr. Puppe also shows this in his report, and both Merkt and Puppe state that the normal N can be resolved into two components, the vertical, which represents the pressure that is sent to the mill screws, and the horizontal, which is the force thrown into the housing posts and creates a bending moment therein.

Mr. Kellar uses the rolling pressure K which he considers applied at the end of a lever whose length is seven-tenths of the horizontal length of contact between the roll and the steel. (See Figure 2, Plate I).

We therefore have supposedly actual horsepowers, as determined by Dr. Puppe's tests, whose magnitudes it should be possible to check with either of the two methods which have been published.

We need not take the time now to go into Mr. Merkt's analysis in detail. You can check through the several formulae at your later convenience. The speaker has done so and found them to be perfectly rational and sees no reason why a force P cannot be used for determining the horsepower rate regardless of whether the normal N does, or does not, bisect the arc ab.

The force P, after all, amounts to a resistance, at the circumference of the rolls, whose magnitude, it might be stated, is the product of the width of the steel in the pass times the reduction taken (this is the area of cross section displaced), and the resistance in pounds per square inch the steel offers to deformation (this is usually taken as the tensile strength of the steel at rolling temperature).

In attempting to check back from Dr. Puppe's table we note that we have only K, the rolling pressure, to begin with, and that K is a function of the normal N whose value we do not know. We however know that H, the resistance at the circumference of one roll, is equal to half the resistance P, and that $N = P\sqrt{\frac{R}{t_1 - t_2}}$. Therefore, by substituting the values of H and N in the formula $K = \sqrt{N^2 - H^2}$ we can solve for P. See Figure 1, Plate I. Then as $H = \frac{P}{2}$ we can, by applying a pull H at the circumference of one roll, obtain the twisting moment in that roll.

In the case of Pass 2, Test 1, H amounts to 50635# from which the twisting moment amounts to 408680 inch pounds.

Checking the twisting moment by Kellar, we apply the force K at distance .7 ad (in this case .7x3.08" or 2.156") and obtain a twisting moment of 550490 inch pounds. (See Figure 2, Plate I).

Since these two moments do not agree it is probable that Kellar's formula, as published, applies only to blooming mills as discussed in his articles.

Assuming now that Dr. Puppe was nearly correct when he selected .075 as the coefficient of neck friction, and that his horsepowers for displacement are therefore quite reliable, we can check for the twisting moment in one roll which corresponds to the energy he obtained for the displacement.

For Pass 2, Test 1, this is given as 1022.9 H.P. sec. (See Line 50, Table No. 1) which when applied in the well known formula $\frac{63025 \text{ H.P.}}{N} = \text{Pr}$ gives us a twisting moment in one roll of $\frac{63025 \times 1022.9}{2 \times .82 \times 119.1} =$ 329375 inch pounds.

Using this moment and calculating the length of the lever arm l_1, we get 1.29" as against 2.156" by Kellar. (Figure 3, Plate I).

We are told that for each force that exists there is always an equal, but opposite force, called a reaction. If that be true, the force K which goes to the screws is opposed by an equal force which is sent to the seat in the housing windows. The force K is made up of the resistance of the steel in pounds per square inch and the area upon which it acts, i.e., length ad times the width of the steel in the pass.

The steel certainly offers the same resistance to the top roll as it does to the one below. Therefore the area of contact with the two rolls must be the same or we have two values for K.

The rolls however usually have different diameters, see Figure 1, Plate II, and if the same bite is taken by each roll, the lengths of contacts cannot be alike. That being true, the area of contacts cannot be equal and the two pressures K cannot balance each other.

There must be, therefore, a condition of roll penetration which will produce the same length of contact between roll and steel. (See Figure 2, Plate II). That length must be the mean proportional to the two parts into which the two roll diameters can be divided.

Referring again to the figure, if we consider the height of the steel before entering the mill of line's thickness and draw the two rolls overlapping each other by the amount of the reduction $t_1 - t_2$, we can calculate the length of contact by the method shown in Figure 3, Plate II.

In the case of Pass 2, Test 1, this is 3.08".

It certainly appears that for all practical purposes it would be accurate enough to scale the length from a full size layout accurately drawn.

If we now use the twisting moment previously obtained for Dr. Puppe's H.P. for displacement in pass 2 (Figure 3, Plate I) and solve for the force H, we get $\frac{329375}{8.071} = 40810\#$.***

***The value 40810# can be checked very closely by the theory of similar triangles. Referring to Figure 4, Plate II, the force H can be calculated from the relation $K : H = \left(R - \frac{5}{16}r\right) : \left(\frac{2}{5}ad\right)$ i.e.

$$255330 : H = \left[8.071 - \left(\frac{5}{16} \times 1.2216\right)\right] : \left(\frac{2}{5} \times 3.08\right)$$

from which H = 40910, or only 100# more than 40810.

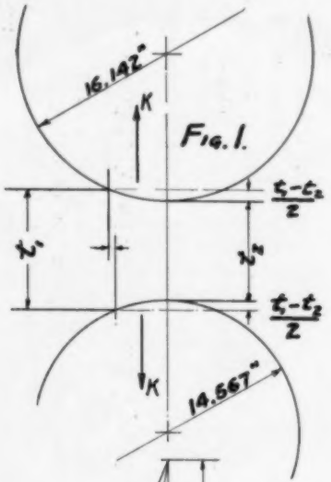


Fig. 1.

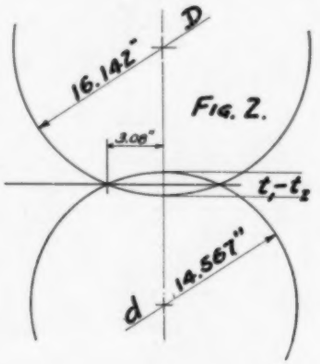
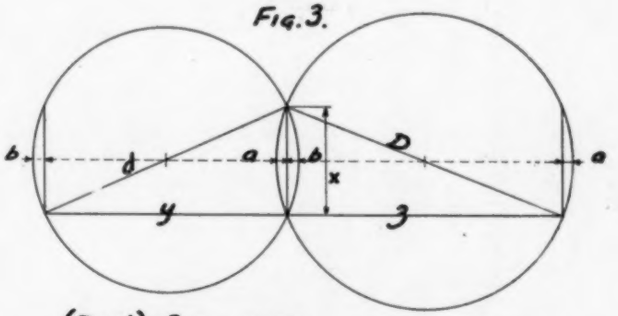


Fig. 2.

Fig. 3.



$(D+d)-2r = y+3$

$d^2-y^2=x^2$, $D^2-3^2=x^2$, $\therefore D^2-d^2=3^2-y^2$

$t_1 = 6.614"$ $\quad t_2 = 5.320"$ $\quad 1.294" = r$

$D^2 = 260.5642$ $\quad d^2 = 212.1975$ $\quad 3^2-y^2 = 48.3667$

$D = 16.142"$ $\quad d = 14.567"$ $\quad 30.709"$ $\quad 2r = 2.588"$ $\quad 28.121"$

$28.121 = y+3$ $\quad y = 28.121-3$ $\quad 3 = 28.121-y$

$28.121^2 = 790.7906$

$48.3667 = 3^2 - 790.7906 + 56.242\,3 - 3^2$

$839.1573 = 56.242\,3$ $\quad 3 = 14.9204"$

$\frac{1.2216"}{16.1420} = 2a$ $\quad a = .6108"$

$(3+a) : \frac{x}{2} = \frac{x}{2} : a$ $\quad \frac{x}{2} = 3.08"$

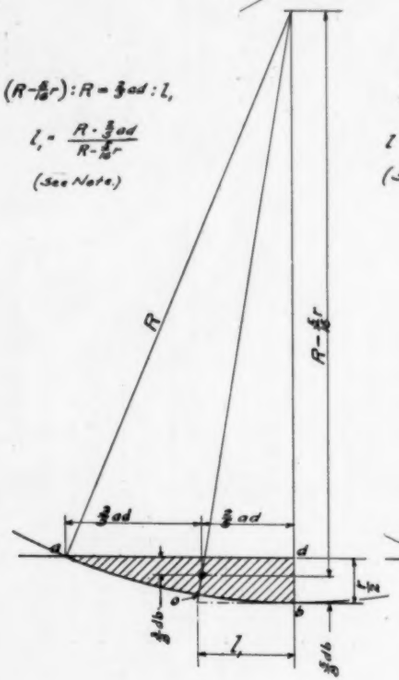


Fig. 4.

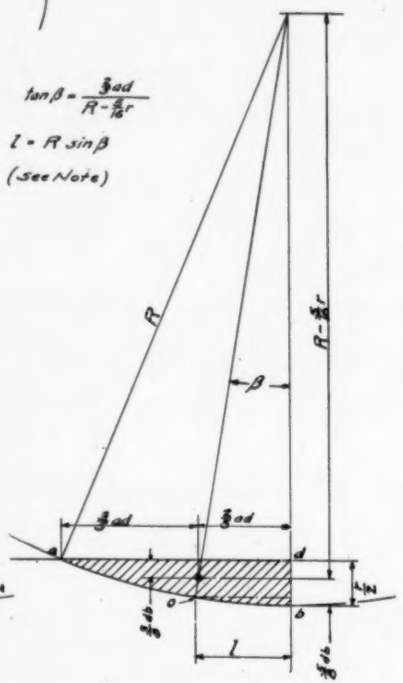


Fig. 5.

Note:—

$ad = \sqrt{R^2-(R-\frac{r}{2})^2}$ when rolls are of same diameter

ad = Mean proportional when roll diameters are different.

r = actual reduction if rolls are of same diameter.

r = equivalent reduction (2a Fig 3) if roll diameters are different.

PLATE II

F. Woldorf, Pittsburgh Dec. 11, 1929.

As this force in the case of Merkt is $\frac{408680}{8.071} =$ 50635# it is possible that the angle his normal makes with the vertical is too great, and possibly the normal does not bisect the arc ab after all.

Suppose, for the purpose of discussion, we assume that the normal N passes through the center of gravity of the area of work. (Shown sectioned in Figures 4 and 5, Plate II). This point is approximately $\frac{3}{5}$ ad back and $\frac{3}{8}$ db down from the point of intersection between the roll and the steel at point of entry.

If we then calculate the length of the lever l_1 by the formula in Figure 4, we get 1.292", or only .002" more than we previously obtained for l_1 for the H.P. of displacement in the table. (See Figure 3, Plate I).

This new length, 1.292", with the rolling pressure K taken from Dr. Puppe's table would give us practically the same horsepower for displacement that he obtained after he deducted the roll neck friction which he calculated on the basis of .075 for the coefficient of friction. See Calculations Pass 2, Plate III.

This however does not prove the correctness of the coefficient .075 and unless it somehow can be checked with some degree of certainty we cannot consider his horsepowers for displacement as final, but only nearly so.

There remains one refinement that can be gone into, which will enable one to approach a check on the correctness of the coefficient of friction used by Dr. Puppe.

If the distance l, Figure 5, Plate II, locating the point O at the circumference of the roll, is calculated, and the horsepower for displacement be calculated from pressure K applied on the end of level l, we should be able to do this.****

The length l for Pass 2, Test 1, figured by trigonometry, is 1.277". The horsepower for displacement, therefore, would be only $\frac{1.277}{1.29}$ of 1247.4 or 1230 H.P. See Plate III.

The difference 1247.4 — 1230 or 17.4 H.P. should then be added to the 412.3 given in the table for roll neck friction and a new coefficient of friction calculated from the result, 412.3 + 17.4 = 429.7 H.P.

This would show $\frac{429.7 \times 550}{2 \times 255330 \times 5.94} = .078$ as the probable coefficient of friction.

The difference would only be $\frac{.078 - .075}{.075} \times 100$ or 4% in the neck friction, while the difference in horsepower for displacement would amount to only $\frac{1247.4 - 1230}{1247.4} \times 100$ or 1.4%.

****In using the formulae in Figures 4 and 5, Plate II, the reduction r is the actual reduction $t_1 - t_0$ if the rolls are of the same diameters. If the roll diameters are not alike, the reduction r should be taken as twice a or b in Figure 3, Plate II, depending upon which roll is used in the calculations.

A hurried check for Pass 4, Test 1, would show the following values:

Length of contact between roll and steel	= 2.8"
Length of lever arm l, Figure 5, Plate II	= 1.1569"
Length of lever arm l_1, Figure 4, Plate II	= 1.17"
Difference between the two lengths l_1 and l	= 1.12% of l_1
Value of H by Merkt	= 33190#
The moment of the rolling pressure K, l_1 = 1.17	= 206230"#
The moment of the rolling pressure K, l = 1.1569"	= 203920"#
The moment by Merkt 33190x7.697"	= 255560"#

The twisting moment corresponding to the horsepower for displacement in Pass 4 as given in Table No. 1 is $\frac{592 \times 63025}{.9 \times 2 \times 115.5} = 179000$"#.

$\frac{592}{.9} = 658$ H.P. See Calculations Pass 4, Plate III.

The horsepower on basis of l = 1.1569" is $\frac{2 \times 203920 \times 115.5}{63025} = 748$ H.P. for the mill.

There is therefore a difference of 748 — 658 or 90 H.P. somewhere.

On checking through all energy calculations for this test it was discovered that the energy given up by the rotating masses is 875 H.P. seconds for pass 4, and 1716 H.P. seconds for pass 6.

Apparently an error of 100 H.P. seconds was discovered for pass 4 by Dr. Puppe's calculators and the wrong figure was changed by mistake by the man who made the entries.

If the corrections are carried through, it will be found that the H.P. seconds for displacement for Pass 4 should be 690.3 instead of 592. Reducing the 690.3 H.P sec. to H.P. we get 767 H.P. for displacement as against 748 as obtained when the lever l = 1.1569 was used.

The difference 767 — 748 or 19 H.P. should therefore be added to the 276.6 H.P. shown for neck friction and the coefficient of friction calculated for that result, i. e., 276.6 + 19 = 295.6 H.P., and the coefficient of friction would figure out as .08.

The correction in the neck friction would therefore be $\frac{.08 - .075}{.075} \times 100 = 6.9\%$ and the H.P. for displacement would change only 2.5%.

The two analyses just made indicate that the coefficient of friction for 12" necks on plain bearings may safely be taken as .07 to .08. The average .075 used by Dr. Puppe was in reality quite correct for the 570 mm stands.

The object in going so thoroughly into the question of Dr. Puppe's horsepower for roll neck friction and displacement was of course that of determining the reliability of those values and then of approximating the savings that could be effected if roller bearings were used throughout in the 570 mm unit.

That being accomplished, Plate IV shows these savings which, you will note, range from 35% to 52% for the six passes.

Table showing HP requirements for Passes 2 & 4 Test No 1.

With coefficient of plain bearing roll neck friction corresponding to the calculated HP for actual displacement of the steel.

		Pass 2	Pass 4
	Weight of billet in pounds.	896.2	896.2
9	Displacement in cubic inches.	492.36	267.3
17	Temperature, °F.	2197.4	2172.2
14	Total Average Rolling Pressure, Pounds.	255330	176265
2	Time of pass in seconds.	.82	.90
4	Average Roll Speed, R.P.M.	119.1	115.5
3	Roll Speed at beginning of pass, R.P.M. n_1	120.3	116.2
	" " " end " " " n_2	118.1	114.8
	n_1^2	14472.09	13502.44
	n_2^2	13947.61	13179.04
	$n_1^2 - n_2^2$	524.48	323.40
25	Energy from Rotating Masses. $2.705(n_1^2 - n_2^2)$ HP sec. (Metric)	1418.0	875.0
24	Average HP from Rotating Masses. (English)	1706.3	958.7
19	" " " Motor "	256.0	338.3
26	" " Rate for the Pass. "	1962.3	1297.0
22+t	HP Rate for light load losses. "	194.8	193.9
44+t	" " Pinions "	105.9	59.7
47	" " Roll Neck Friction. Coef. frict. .076 "	417.3	276.6
50+t	" " Displacement "	1247.4	767.0
26	Total HP for the Pass. "	1960.4	1297.2
	Approximated length of arm l inches.	1.2769"	1.1569"
	HP Rate for Displacement based on arm length l "	1230.0	748.0
	Difference in HP Rates for displacement, HP and %	17 HP 1.4%	19 HP 2.5%
	Corrected HP rate for Roll Neck Friction "	429.7	295.6
	Difference in HP rates for Roll Neck Friction HP and %	17 HP 4.2%	19 HP 6.9%
	Coef. frict. corresponding to corrected Roll Neck Friction HP rate.	.078	.08

F. Woldorf, Pittsburgh, Pa
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PLATE III.

Pass 2 — $\frac{255330 \times 2\pi \times 1.292 \times 119.1}{12 \times 33000} = 623.7$ HP per roll. $\quad 2 \times 623.7 \times .82 = 1022.9$ HP sec $\quad \frac{1.2769}{1.292} \times 1247.4 = 1230$

$\frac{2}{1247.4}$ HP for Disp'mt.

Pass 4 — $\frac{592 \times 63025}{.9 \times 2 \times 115.5} = 179000$ "# $\quad \frac{592}{.9} = 658$ HP $\quad \frac{2 \times 176265 \times 1.1569 \times 115.5}{63025} = 748$ HP $\quad 748 - 658 = 90$ HP difference.

$\frac{690.3}{.9} = 767$ HP $\quad 767 - 748 = 19$ HP $\quad 276.6 + 19 = 295.6$ HP $\quad \frac{295.6 \times 550}{2 \times 176265 \times 5.75} = .08$

Test No 1. 570 mm. (22.44") 3 High Mill. Roughing Stand.
Rolling 50 × 50 mm (1.97" × 1.97") Billets from a 406.6 kg. (892.616) billet.
Neck diameters, Plain Bearing Mill, 290 mm (11.42").
Neck diameters, Roller Bearing Mill, 12". M.P.D. R.C.C. = 15"

Approximate Power Savings To be expected from Roller Bearings.

		Pass 2. Bearings.		Pass 4. Bearings.		Pass 6. Bearings.		Pass 8. Bearings.		Pass 10. Bearings.		Pass 12. Bearings.	
		Plain	Roller.	Plain	Roller.	Plain	Roller.	Plain	Roller.	Plain	Roller.	Plain	Roller.
Light Load Losses.	(22)	194	10	194	10	194	10	194	10	194	10	194	10
Pinion Stand Losses.	(49)	106	6	60	3	106	6	90	5	71	4	40	2
Roll Neck Friction Losses.	(47)	412	15	277	10	347	13	391	15	342	13	244	9
Horsepower for Actual Displacement	(50)	1250	1250	767	767	1212	1212	1023	1023	770	770	389	389
Plain Bearing H.P. demand, excluding motor losses.	(26)	1962		1298		1859		1698		1377		867	
H.P. demand for Roller Bearing equipped unit.			1281		790		1241		1053		797		410
% Plain Bearing H.P. demand required by Roller Bearings.		65		61		67		62		58		48	
% " " " " saved " " "		35		39		33		38		42		52	

All decimals omitted

F. Weldorf, Pittsburgh, Pa. Dec. 30, 1929.

Coefficients of Friction assumed.
For Roll Necks taking the Rolling Pressures. . . . . .075 (Plain bgs) .0021 (Roller bgs)
" Roll Necks under light load, and pinion necks. . . . .107 (" ") .0042 (" ")

PLATE IV.

The bearing in this case is the same size (12"x17½") as has been in use for some time in our own 22" mills at Canton, Ohio. These bearings have been subject to over-load continually and as the over-load condition in the 570 mm mill would not exceed 23% for 8-1/3 seconds or 30% for 3½ seconds this bearing would answer in every respect.

The coefficient of friction of the 12"x17½" Timken bearing, referred to its M.P.D. (15") of the roller center circle, is taken at .0021 for the necks that are loaded, and at .0042 for the pinion and light mill necks.

The coefficient of friction assumed for the loaded necks in the 570 mm plain bearing mill has been kept at .075 for the loaded necks. Checks made to approximate the coefficient for the pinion and light necks in this unit show that it amounts to about .107.

Since the torque in pounds feet required to overcome the friction of a journal is equal to $\frac{Wfr}{12}$, in which W is the load on the journal in pounds; f, the coefficient of friction; and r, the radius of the shaft in inches, the ratio of the torques required for roller and plain bearing journals respectively would be $f_r\ r_r : f_p\ r_p$.

On this basis the horsepowers for the loaded necks in the roller bearing mill would be only $\frac{.0021 \times 7.5}{.075 \times 5.71} \times 100$ or 3.7% of those given in Dr. Puppe's tables. The pinion and light load losses would be reduced to $\frac{.0042 \times 7.5}{.107 \times 5.71} \times 100$ or 5.2% of his plain bearing values.

Plate IV shows the savings that could be effected for the six passes in the 570 mm mill if roller bearings were installed on all necks.

It can be clearly seen that the savings for the pinion and light mill necks would amount to considerable and would of course be added to that effected by the bearings on the two rolls that do the actual rolling of the steel.

We recently had an unlooked for demonstration of the efficiency of our bearings on roll necks. Several rolls weighing about 26000# each were removed from a four high mill for the purpose of inspecting our 18" bearings. No effort was made to level up the boxes which were merely set on skids on a wood block floor where they remained for four days before we arrived at the plant. The grease had ample time to become cool and stiff during that interval.

Upon attempting to back out a ¾" locking screw from the locking collar on one neck, the pull on the wrench rotated the roll which had to be spragged before we could turn the bolt.

We thought this a rather unexpected, but interesting demonstration on the part of our bearing and tried to rotate the roll further. We found we could keep it moving by pulling on the wrench, and further, that one man could rotate the roll by merely gripping the wabbler flutes with his bare hands and pulling on same.

The mean pitch diameter of the roller center circle of our bearing is about 24". Assuming a coefficient of rest for that size bearing of .0042 the pull on the end of the wrench, on say a 20" leverage, would be $\frac{26000 \times .0042 \times 12}{20} = 66\#$.

As the operator did not brace himself in order to increase the effectiveness of his pull, it is quite evident that the pull required to rotate the roll was less than 66#.

With a demonstration of that kind to report, there should be no hesitancy on your part about accepting the savings shown on Plate IV as being en-

tirely within the bounds of possibility if roller bearings were put on the roll and pinion necks of the 570 mm unit.

In conclusion the speaker can only repeat what he said at the beginning of the paper. It can truthfully be said that a thorough study of the applicability of roller bearings to steel mill equipment cannot help but inject new interest and enthusiasm into the daily tasks of the steel mill engineer.

The economies which will result from their liberal use will in time, not only be measured in so many dollars and cents saved for the mill owner, but they will so reduce the time devoted to maintenance and renewals that next to the slogan "Safety First," "Roller Bearings" will become the watchword whenever new equipment is about to be purchased, or old equipment is torn down for renewal or repair.

Roller Bearings are here to stay, and he who delays looking for uses to which they can be applied is overlooking one of the greatest boons that has ever been introduced into the engineering, the construction and the operation of a steel plant.

Recent Developments in Dynamic Braking Lowering

By D. C. WRIGHT*

Within the last few months improvements in circuits of dynamic braking lowering controllers have been made, which give four distinct advantages over the connections that were formerly used. In order to full understand these advantages and the scheme of connections, it might be well, first, to review the circuits which have been used for the last few years.

Prior to 1910, various attempts were made to use dynamic braking instead of mechanical load brakes and other friction devices, then in common use on hoists for retarding the load in the lowering direction. Although these attempts were partially successful on special hoists where the load was always sufficient to overhaul the mechanism and where shunt wound magnetic holding brakes could be used, they were not successful on standard E.O.T. cranes where the load was liable to vary from a positive or driven load of 25% of full load, to a negative, or overhauling, load of more than 100%, and where series wound holding brakes had to be used for safety reasons. Under these varying load conditions, the motor at times had to deliver a driving torque and at other times a retarding torque. To operate successfully the motor should adjust itself to these varying conditions without thought or manipulation by the operator.

About 1910, certain controller connections were devised to give the motor the ability to adjust itself to the load, and since that time all of the controller manufacturers have used practically the same connections; the only difference being in the method of acceleration and the way in which the various resistance banks were inserted or cut out. These connections were such that the series motor was given shunt motor characteristics in the lowering direction, by separately exciting the series field from the line and connecting the armature so that it also received current from the line if load had to be driven down; or, if the load was sufficient to turn the armature, it generated current as a series generator to supply the field excitation or at faster speeds to pump current back into the line.

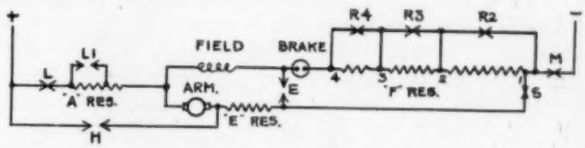

Fig. 1. First point lowering—Old scheme of connection.

Figure 1 is a typical connection diagram for a magnetic dynamic braking controller and shows the actual connections to the motor on the first point lowering. Contactors, L, M, S, R2, R3 and R4 are closed and contactor E, which is magnetically opened and spring-closed, is open. It will be noted that the line current passes through the limiting resistance A and then divides, part going through the series field and series brake, and part going through the armature and resistance E. The series brake is released by the current passing through that branch of the circuit. If the load on the motor is light and has to be driven down, the motor acts as a shunt motor and supplies the necessary torque, driving the load at a slow speed. If the load is heavy and overhauls the motor, then the armature generates a retarding torque by the generated current which passes through the field, series brake, the resistance E, and back to the armature. This dynamic braking circuit of low resistance prevents the motor from exceeding a slow speed.

Succeeding points on the master controller open contactors R4, R3 and R2 and close L1, so that on the last point we have all of resistance F in series with the field and most of resistance A cut out of the circuit. Light or driven loads will be lowered faster because of two conditions; first, because of the weaker field strength and second, because of the higher voltage impressed on the armature due to cutting out of the A bank. Heavy loads will lower faster because of the higher resistance in the dynamic braking circuit. The series brake has to release on

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the first point in order to get any movement of the armature. If it does not release on the first point it will not release on any of the succeeding points, as each step inserts additional resistance in the brake circuit. Therefore, in order to insure positive operation of the brake all controller manufacturers allow about 150% full load current to flow from the line on the first point. And even with this amount of current it is frequently necessary for the operator to hesitate on the first point before proceeding to the other points, especially if the brake is a little out of proper adjustment. It will be noted that, if the resistance E could be increased in value, more current from the line could be forced through the brake circuit. However, E cannot be made of a high value or else heavy loads would lower too fast on the first point, and, in addition, the higher E is in resistance, the slower the speed will be when lowering light loads on the last point, due to less impressed voltage on the armature. Another factor in the ability to release the brake on the first point is the unknown resistance in the divided circuit. The trolley bars and collector shoes may be long or in poor surface condition, so that, even with 150% current from the line not more than 50% current may pass through the brake.

Some schemes cut out all of the resistance bank A upon closure of contactor L1 in order to secure a higher speed on the last point with light loads. This, however, is likely to give a high peak of current and open the circuit breaker if the master is moved rapidly from the first to the last point. The motor will not have had a chance to generate much C.E. M.F. and the only resistance in the armature circuit is the E bank. For this reason, other schemes leave part of the A bank permanently in the circuit. In order to increase the speed of light loads, some schemes insert additional resistance in the F bank to weaken the field to a further extent on the last point. This will tend to allow heavy loads to lower at a dangerous speed, and to prevent this, load relays connected in the line circuit are used to prevent insertion of the extra resistance on certain loads above predetermined values. A better way to increase the light load speed is to cut out part of the E bank on the last point so as to increase the voltage applied to the armature and, at the same time, insert the same amount of resistance in the F bank.

Realizing the difficulties in releasing the series brake, attempts have been made to connect the brake in some other portion of the circuit. This cannot be done, however, as it is necessary that the brake be in the dynamic braking circuit on the last point; otherwise, the brake is liable to set when lowering certain loads. At about 75% full load, the motor is operating at a speed sufficient to generate full voltage, and the current taken from the line is zero. At heavier loads the motor generates higher than line voltage, and at this point starts to pump current back into the line. Therefore, the flow of current outside the dynamic braking circuit reverses. Also, the current in the armature circuit reverses, depending on the load. For these reasons the only place for the brake is in the field circuit.

It will be noted that, in order to get any movement of the armature at all, it is necessary to operate seven contactors and release the brake. For this reason, it is frequently difficult to get short, accurate movements of the armature. It is hard for the operator to judge how long to leave his master on to get a short movement of the crane hook, and, if he leaves the master on too long, he gets a considerable movement, especially with a heavy load.

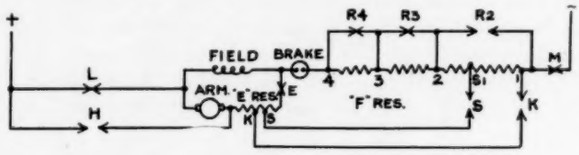

Fig. 2. First point lowering—New scheme of connection.

Recognizing the two limitations of magnetic dynamic lowering controllers, that is, the difficulty in releasing the brake in the lowering direction and the difficulty in making short accurate movements, a new scheme of connections has been devised as shown in Fig. 2. On the first lowering point contactors L, M, R3 and R4 are closed and the spring closing contactor E is left closed. The line limiting resistance is that portion of the hoisting resistance between 1 and 2. The motor connections differ from this in Fig. 1, in that the armature and resistance E parallel the field only, instead of paralleling the field and brake. All of the current taken from the line, instead of only a part of it, flows through the series brake. The resistance 1-2 is about one-half of the total hoisting resistance and allows full load current through the brake, whereas, with other exciting schemes only one-half full load current generally flows through this circuit with 150% full load taken from the line. By putting twice the amount of current through the brake it releases in about one-fourth the time. The brake is, therefore, released positively and quickly on the first point even if considerably out of adjustment. On heavy overhauling loads, the dynamic circuit includes the armature, field and resistance E and not the brake, so that, heavy loads are retarded to a slower and safer speed than with the other schemes of connection. On light loads the armature has a lower voltage impressed on it and a slower speed is obtained on the first point.

It will be noted, that, in order to get the brake released and the motor started, it is necessary to operate only four contactors as against seven in the other schemes of connections, and also that full load current is obtained, instead of about half load to release the brake. It is readily seen that the operator can make short accurate movements much more easily. Even if the master is not left on long enough to close R3 and R4, half load current is still obtainable through the brake. In fact, under these conditions slightly more current would flow through the brake than on the first point hoisting.

On the second point lowering contactor S is closed and E opened. This weakens the field strength and increases the voltage applied to the armature, thus giving higher speeds when driving down light loads instead of slower speeds, as is obtained on some schemes of connection for dynamic lowering controllers. The resistance between 2 and S1 is higher than between E and S. Therefore, on the second point, the resistance in the dynamic cir-

cuit is greater and overhauling loads will descend faster.

On the third point, contactor R4 is opened, increasing the resistance in the field circuit, giving higher speeds on all loads . The fourth point opens R3, again giving higher speeds due to weaker field strength.

On the fifth and last point, K is closed and S opened. This weakens the field strength and impresses a higher voltage on the armature, giving a higher speed to drive down light loads. It also gives a higher resistance in the dynamic circuit, thus increasing speeds of overhauling loads.

Comparing the last point connections of Fig. 2 with those of Fig. 1, it will be seen that Fig. 2 has no external resistance in the circuit and that most of the E bank has been cut out of the armature circuit. If the resistance, 1 to 4, is the same in both cases, then Fig. 2 will give much higher speeds on light loads—a feature much desired. Also, since the resistance, E to K, is shorted out of the dynamic circuit, the speed in lowering overhauling loads with Fig. 2 will be less—a feature desired particularly when handling heavy overloads. With the older schemes of connections, the speed of lowering empty hook was about 150% of the rated full loads speed of the motor (this, of course, depends considerably on the friction load of each installation). With Fig. 2 connections it is easily possible to secure 175% to 180% of rated full load speed, or about a 20% increase in speed over Fig. 1. The older connections gave about 250% of rated full load speed when lowering full load on the hook. Although this is not a dangerous speed on moderate or slow speed motors, it is dangerous on high speed motors, and would also reach dangerous speeds on slow speed motors as soon as the load exceeds full load to any extent. Fig. 2 connections reduce the full load lowering speed to about 200% to 210% of rated full load speed. By properly proportioning the resistance, both the no load speed and the full load speed of Fig. 2 connections could be increased.

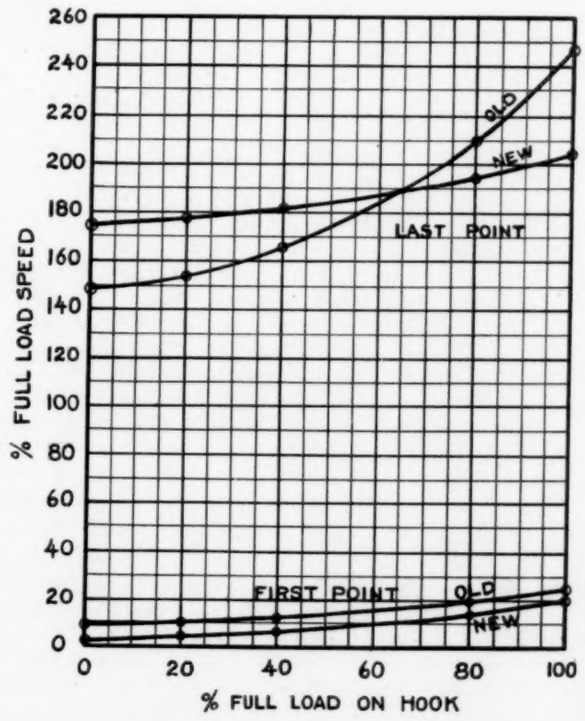

Fig. 3. Lowering speeds for various loads.

Fig. 3 shows speed-load curves on a 15-ton crane which was first equipped with a controller as shown in Fig. 1 and then changed to Fig. 2. The first-point speeds of all loads are slower and safer with the new controller, and the light load speed on the last point considerably higher.

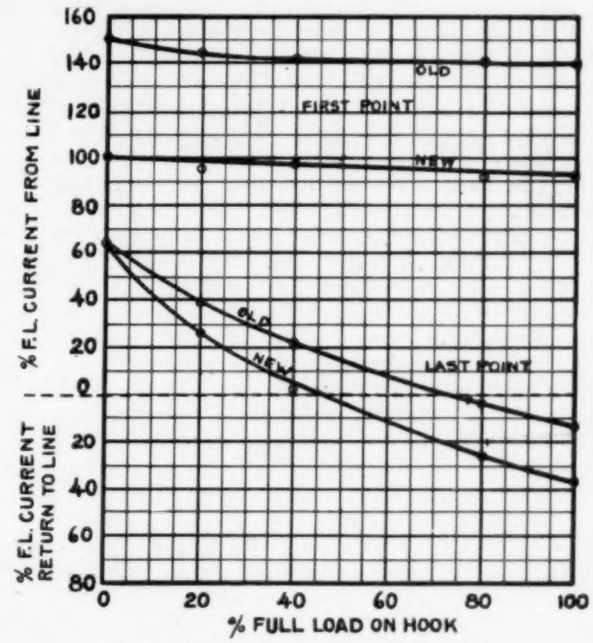

Fig. 4. Line current at various loads.

Fig. 4 shows current taken from the line and returned to the line for various loads. With the old controller, loads above 75% full load gave a power return; whereas, with the new controller loads above 50% gave power return. The current on the first point with the new controller averaged only about 66% of that with the old one.

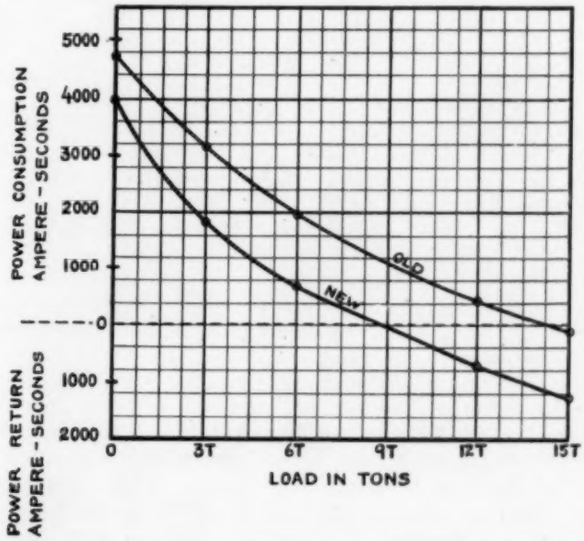

Fig 5. Power required to lower various loads 22 feet.

In order to show more clearly the saving in power afforded by this type of controller, the curves in Fig. 5 were calculated. With the old controller it required 60 seconds to lower the empty hook through 22 feet; and if the operator hesitated 2 seconds on each acceleration point, the total power consumption would be 4776 amp.-sec. The new controller would lower the empty hook through the 22 feet in 51 seconds and at a lesser amount of current; or the power consumption would be 3989 amp.-sec. or a saving of 19% in power. Assuming the same acceleration time, and a corresponding decreased operating time due to increased speeds for heavier loads, the curves show a saving of 42% in power to lower the 3-ton load, 60% for the 6-ton load, and 295% for the 12-ton load. That is, on the 12-ton load the old controller actually took 472 amp.-sec. from the line; whereas, the new one put back 921 amp.-sec. more than it took. On the 15-ton load both controllers put more current back into the line than they drew, but the new one put back 12 times as much as the old one. These amounts of power on a 25 H.P. hoist may seem small, but over a years time and on a large number of controllers it may represent several thousand dollars saving.

As stated in the preamble, and as shown by the curves and diagrams, these new circuits give four distinct advantages over the older schemes, as follows:

First: Ability to release the brake in the lowering without making it necessary for the operator to hesitate on the first point; and in fact, allowing the operator to throw the master rapidly to the last point and secure the maximum possible rate of acceleration without drawing excessive current from the line.

Second: An increase in lowering speeds of light loads without increasing the speeds of heavy loads and without additional apparatus or complications.

Third: An increase in accuracy of control, by making it possible to secure short accurate movements of the armature under all conditions of load —this feature so highly desired on foundry cranes for setting molds and flasks, on riveter cranes, and when setting rolls, foundation and machinery in place.

Fourth: A material saving in power due to less current consumption and faster speed.

Further advantages resulting from the above are: Lower voltage drop, due to lower peaks of current—lower heating on the motor—less contactor wear, due to lower current and a lesser amount of inching for accurate control.

Gearless Speed Reducers

By H. M. EDMUNDS*

The type of speed changer which is the subject of this lecture has been given the classification of "friction type." The word friction is perhaps unfortunate in a class of machinery in which no rubbing or sliding occurs between the surfaces of contact. The name, however, is derived from an age old form in which the rim of one disc presses the surface of another and in which the power transmitted is strictly limited and the efficiency relatively low. Such devices have the advantage of giving variability of ratio by sliding one disc relatively to the other. They are employed in presses for coin and metal making and a number of similar uses. There have also been a number of friction drives developed for light automobiles, but to the best of my belief these have never proved successful in practice.

It was a bold conception, therefore, that the principle of adhesion of surfaces in contact could be utilized for the transformation of power from one speed to another in the multifarious uses for which tooth gearing, belts, chains and worm wheel drives have hitherto been used.

It is appropriate to render to Mr. C. G. Garrard, the inventor of the device I am to describe in this lecture, a recognition for his skill in seeing the possibilities and designing a machine which has proved itself of great practical utility and to possess certain unique advantages over other methods.

*Crocker Wheeler Electric Mfg. Co., Ampere, N. J.

Two distinct types of speed changer have been developed and I propose now to describe in detail these mechanisms and then to consider their properties, applications and certain technical features. Before proceeding, however, to a detailed description of these two types. I would like to say a few words about the difficulties which the inventor set out to overcome. In the first place, if we consider the case of driving a smooth cylindrical roller made of hard steel and ground to a true surface by another one of different diameter pressed against it with sufficient force to drive without slipping, we are faced with the fact that under normal conditions, the bearings which support these two rollers would have themselves to transmit this force of adhesion. This would obviously be a grave limitation as the forces involved are relatively high and the bearings, to resist such forces, would have to be large and costly. It will be seen that in the two constructions to be described, this difficulty has been overcome.

Another feature which is of great importance in this type of speed changer is that the pressure necessary to transmit the drive should vary in proportion to the load to be transmitted. In other words, the very great pressure necessary for the extremest overloads should not have to be borne when the load is reduced, as this would clearly curtail the life of the mechanism and also impair the efficiency of light loads. We may draw a useful analogy to this phase of the problem from the case of the locomotive driv-

ing a train along steel rails, in which the driving force depends upon the adhesion between wheel and rail to accelerate the train and to transmit the draw bar pull necessary to overcome the frictions of running which the speed requires. In the case of a locomotive, the weight is constant and if the tractive effort exceeds a certain proportion of this weight, slipping occurs and it is necessary to resort to sanding or other means to obtain sufficient adhesion for driving. In the design of the two types of speed changer we are to consider, the force of adhesion is automatically altered to suit the load.

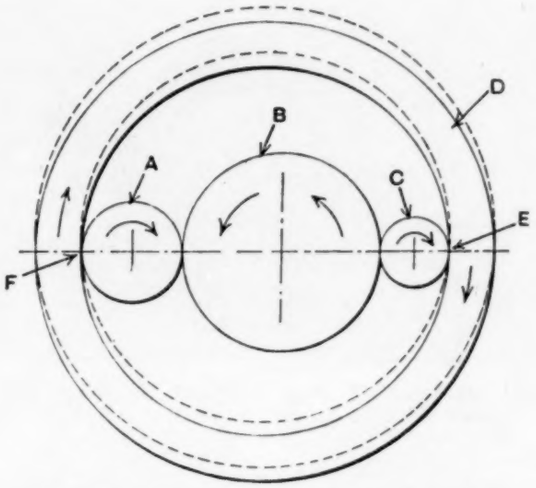

FIG. 1—Diagrammatic View of Cylindrical Type Speed Changer.

In Fig. 1, we have a diagrammatic view of what is called the C type speed changer. The letter "C" in this connection stands for cylindrical. The rollers in this type are of cylindrical form ground as perfectly as is possible and made of a high quality of chrome steel, heat treated to obtain a surface hardness about the same as that which is employed in standard ball and roller bearing practice. The three rollers are shown in end view in the illustration and are marked "A", "B" and "C". "A" and "B" are called the high speed and low speed rollers, one of them being connected to the driving motor and the other to the machine to be driven. In the case of a reducer "A" is connected to the motor and "B" to the machine. "C" is called the idler and is put there for the purpose of enabling the ring marked "D" to rotate with the other three rollers. If only two rollers were employed, the outer surfaces would either be moving downwards or upwards at the same time and the ring placed over them would have no effect but to prevent rotation. The ring "D" when placed over the three rollers is free to turn when the roller "A" is rotated. The surfaces, that is the external surface of the three rollers and the internal surface of the ring, all move with the same velocity and this is true irrespective of the diameters of rollers and ring. The two outer rollers "A" and "C" are in practice provided with flanges at both ends of their working surfaces so as to retain the ring in its correct position, but in other respects the ring is completely floating and is supported only by its two contacts with the rollers "A" and "C". The interior surface of the ring is also ground to pure cylindrical form and is made a few thousands of an inch smaller than the sum of the diameters of the three rollers. The reason for this is to give it a little initial fit or compression not enough to transmit the full load torque of the reducer, but to insure that at all times there will be a compression exerted by the ring upon the three rollers.

Let us consider now what will happen if we place bearings on shafts through each of the three rollers but fix the shaft to the roller "B" so that it would be impossible to turn it. If we then apply a turning force to the roller "A" it will be clear that since "B" cannot move there will be slipping at the surface between "A" and "B". The idler "C" on the other hand will not turn since "B" remains stationary and the ring will in consequence move out of its center position into the position indicated by dotted lines. It should be noted that in this position the line joining the outer contacts between ring and rollers will no longer be a diameter of the ring. In mathematical language, the ring contacts are no longer at the ends of a diameter but are at the ends of a chord. This chord is, however, stretched by the solid mass of the three rollers to the same length as was previously occupied by the diameter. This involves an increase in the compression exerted by the ring on the three rollers and it is this action upon which we depend to obtain the adhesive force necessary to drive under all conditions of load.

The small model illustrates this phenomenon. If I hold the central shaft steady and try and turn the driving shaft, the ring moves slightly until the binding force becomes too strong for further turning. Exactly the same thing happens if the machine is running at speed. Any increasing torque displaces the ring from its central position.

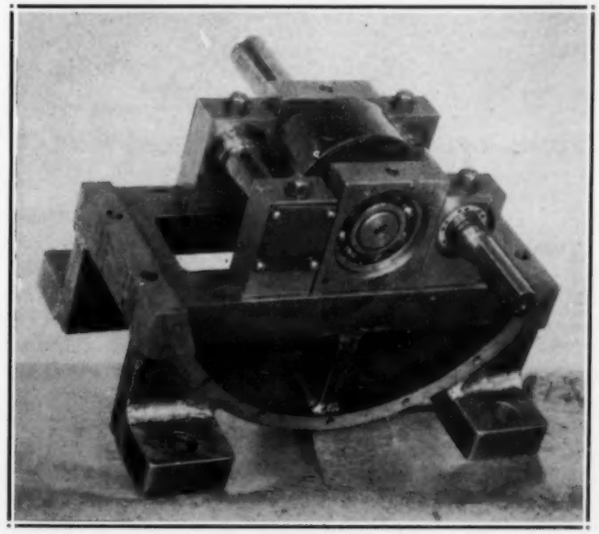

FIG. 2—Cylindrical Type Speed Changer With Binding Ring Removed.

In certain tests which we have carried out on these, we placed a distance indicator in the housing at the top of the speed changer so as to measure the height of the top of the ring at varied loads and found that in the case of a ring of about 10" external diameter, displacement of as much as ⅛" took

place at full load. Of course ⅛" represents a very small extension of the ring, but it is our practice to make these rings of very considerable strength, the external diameter being 1½" times the internal, as this has been found in practice to be an important feature in the design to secure great overload capacity and resistance to shock loads.

The photographs illustrating the mechanical construction of the two types of speed changer show the method by which the bearings are supported upon a firm base which we call the bridge. The bearing blocks, which are placed on this bridge, have a certain amount of latitude between the bolts which hold them and the holes bored in the bridge. By this means we are able to set the three rollers exactly in the position which they should occupy and clamp them there. It will be noted that the low speed roller is placed in the center of the housing, while the high speed is to one side. We make use

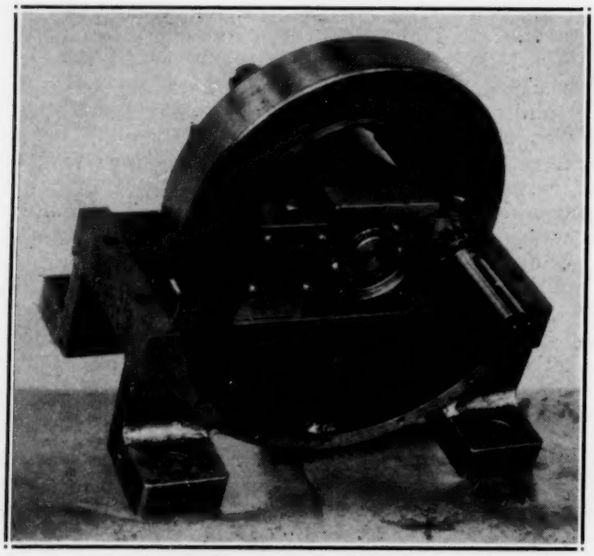

FIG. 3—Cylindrical Type Speed Changer, Showing Rollers and Binding Ring.

of this fact in our speed increasers as it is possible to mount the housing upon a special rear shield and thus to build in the speed increaser with the electric motor. By this means we dispense with base and coupling and provide a self contained unit for which the motor runs at its normal speed generally of 1750 or 1150-rpm and the high speed shaft delivers the power at any speed which may be wanted. As a reducer, the C type is employed only as an independent speed changer and is mounted to its motor on a bed plate and coupled to it by a flexible coupling. The firm of Frederick Krupp of Germany, who own the license for the Garrard inventions for that country, have built a large number of these C type speed changers for powers up to 200-HP and for a great variety of purposes. This is interesting as this firm has a large gear department and manufactures herringbone speed changers and are therefore in a position to judge of the relative merits of adhesion type and of tooth gearing.

The S or spherical type speed reducer consists of the following elements: A central shaft of cylindrical form surrounded by three rollers which have a conical portion in the middle and are of cylindrical form at the end. The conical portion consists of two opposed cones making a small angle with the axis. These three rollers are mounted in what we call a cage, which is a cast iron structure which constitutes the slow speed member of the reducer. The three rollers also contact with a pair of rings which remain stationary and are held in the housing by three hardened steel pins which engage with the square openings provided in the side of one of the rings.

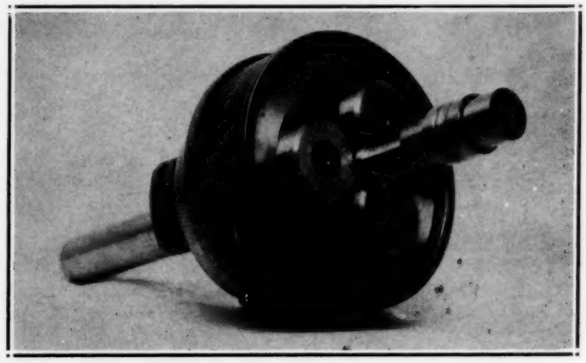

FIG. 4—Spherical Type Reduced, Showing High Speed Shaft in Center, Low Speed Rollers and Stationary Rings.

The two rings are provided with a series of inclined surfaces which engage with each other in such a manner that if one is rotated relatively to the other the inclined surfaces slide upon each other bringing the two rings further apart. In this manner we have a similar action to the one already described for the C type speed changer. When extra torque is applied one of the rings moves round and so provides the additional force of compression to enable the speed reducer to work without slipping and when this extra torque is removed the free ring slides back and reduces the compression. In this case we have what is really a planetary system which

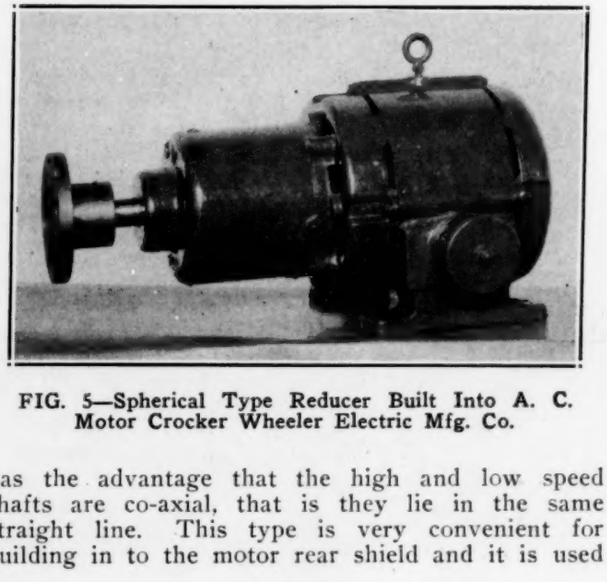

FIG. 5—Spherical Type Reducer Built Into A. C. Motor Crocker Wheeler Electric Mfg. Co.

has the advantage that the high and low speed shafts are co-axial, that is they lie in the same straight line. This type is very convenient for building in to the motor rear shield and it is used

for motor-reducer combinations where low speeds are needed. The different standard sizes all have a fixed size pair of rings and ratios between 10:1 and 4:1 are obtained by varying the size of center shaft and rollers. The cage remains the same so that it is our practice to produce these speed changers complete with the exception of shafts and rollers, which are produced when the required ratio is known. By this means, any ratio can be supplied and we are not restricted to a fixed series of ratios as is the case with tooth gearing.

It will scarcely be necessary to explain to a group of electrical engineers that the horsepower which can be obtained from any given frame size of motor decreases very rapidly as the speeds decrease, so that it is clearly of great advantage to be able to use a motor of say 1150-rpm combined with a speed changer for low speed drive. Many fan drives come in this category and the saving in material, in space and in cost of a speed reducer-motor combination over low speed motor is very marked. Furthermore, the electrical performances of the motor is much better at the higher speeds. This is true in direct current, still more so in alternating current practice where the power factor of the slow speed motor is a serious objection to its use.

Gearless Speed Reducers

By H. M. EDMUNDS

Discussion

Discussed by

A. L. Reichert, Electrical Superintendent, Bourne Fuller Co., Cleveland, Ohio.
H. M. Edmunds, Crocker Wheeler Electric Mfg. Co., Ampere, N. J.
A. F. Jones, Chief Electrician, Champion Machine & Forging Co., Cleveland, Ohio.
J. H. Hall, Electric Controller & Mfg. Co., Cleveland, Ohio.
G. A. Kruse, Mechanical Engineer, Hospital Specialty Co., Cleveland Ohio.
J. R. Lewis, Central Sales Mgr., Crocker Wheeler Electric Mfg. Co., Pittsburgh, Pa.
J. L. Finnicum, Sales Engineer, Allen-Bradley Co., Cleveland, Ohio.
A. Oslin, American Steel & Wire Company, Cleveland, Ohio.
J. B. Mason, Sales Engineer, Allen-Bradley Co., Cleveland, Ohio.
S. L. Henderson, Electrical Superintendent, American Steel & Wire Co., Central Furnaces, Cleveland, O.
F. C. Eckworth, Sales Engineer, Crocker Wheeler Electric Mfg. Co., Cleveland, Ohio.
Mr. Beck, Sales Engineer, Ideal Electric Mfg. Co., Cleveland, Ohio.
H. W. Eastwood, Sales Engineer, Cutler Hammer Mfg. Co., Cleveland, Ohio.
G. H. Gildersleeve, Sales Engineer, I. T. E. Circuit Breaker Co., Cleveland, Ohio.

A. L. Reichert: You have all heard Mr. Edmunds' description of his speed changer, undoubtedly we all have some ideas, some questions we might want to ask him. I myself have received a slightly different idea of the speed changer than I had at first. When Mr. Edmunds was speaking I wondered if there was any way of lubricating, but I see he has already answered that question by saying that it runs in oil.

But there is one thing that in my opinion might be a slight handicap, or I might hesitate a little bit in ordering a speed changer, and that would be because you might be tying up a motor with some kind of expensive equipment, or somewhat expensive and complicated equipment; if anything should happen to the motor, this speed changer might be, as Mr. Edmunds says, fool proof, but I haven't found any motor yet that you don't at some time or other have some trouble with it.

My point is, how much trouble would it be to divorce the motor from the speed changer when the time came to make necessary repairs, or whatever you might want to do, clean the motor or speed changer.

H. M. Edmunds: Well, I think I can deal with this particular question. Of course it is a fact that any one who uses one of these motors and speed changers combined has two things which might give him trouble and might go wrong instead of only one, and if trouble occurs on either of those two elements he has both the elements out of action. That is quite true. The difficulty of divorcing the two, of taking away the speed changer from the motor is not very great. The little nut that I think you have seen in the end of the shaft on the pictures and which is present here in this model, should be removed, then this rod which runs through here can be taken away, together with the four bolts which hold the speed changer on. We can remove the speed changer in a very few minutes from the motor itself.

Of course there is the fact that the speed changer, if the motor has gone wrong will be out of action until the motor comes back, but that would be true of a coupled job. The only argument is really that what one has to do to get these apart is perhaps rather more than what one would have to do in unshifting a coupling. The motor is to this extent non-standard that it has a special rear shield; one, you see, has to have a policy in a thing like this and assume and design it so that it shall not go wrong, and shall not give trouble—everybody is go-

ing to have trouble at some time or other, we cannot be perfectly certain that it will never occur, and I won't pretend that this device is absolutely free from all possibility of trouble. I think that no engineer would expect that, that is not possible, we can only reduce it to the minimum. Sometimes one gets defective steel, a cracked roller; we do our utmost to prevent it, just as the Ball and Roller Bearing people do, and we have an extremely small number of cases of trouble of that kind, but no one would pretend that it is absolutely impossible. However, I do not think that constitutes a serious objection to the combination of this speed reducer with the motor; personally I think in practical working, the conveniences far outweigh the possible limitation which has been pointed out.

A. L. Reichert: What are the practical ratio limits?

H. M. Edmunds: Between 4 to 1 and 10 to 1 in the reducer, and in the C type it can be 10 to 1 down to anything.

A. L. Reichert: Are enormous stresses set up in that under heavy loads?

H. M. Edmunds: Stresses are heavy, but nothing to what the ring can stand. The proportion of what it can stand should be enormous over what it should have to bear, many hundred per cent greater; before you break that ring the force applied would be much greater than any force you could possibly meet with for which the machine is designed.

A. L. Reichert: I was thinking more of distorting the ring.

H. M. Edmunds: It does distort slightly.

A. L. Reichert: I mean permanent distortion.

H. M. Edmunds: The answer really is this: If we took a machine like this, a 2 n.p., and deliberately put it on to 10 h.p. load, it would have a comparatively short life. The surfaces would undoubtedly disintegrate. The answer to that is that it shouldn't be so used. These machines are very strong in their resistance to overload, they stand a great deal; we are producing a new type of speed changer so we get all the bad problems other people have had trouble with, and this is a very good machine for standing up under those conditions.

A. F. Jones: How does the efficiency compare with one of worm gear type?

H. M. Edmunds: The efficiency of these two devices is well over 98%. The worm gear efficiency is a much debated question. I have heard high figures quoted and relatively low ones, it depends very much on the ratio; I think it is much more efficient than a worm drive for the same ratio. This is a 10 to 1 drive and note the ease with which you can turn the shaft from the low speed end. I cannot imagine a 10 to 1 worm gear in which you could even turn it at all from the low speed end, but this turns quite easily.

J. H. Hall: How much does that efficiency change with twice the capacity?

H. M. Edmunds: Ordinary load efficiency you mean, 200%?

J. H. Hall: Yes.

H. M. Edmunds: We have measured efficiency up to 150% load with no falling off, it was still 98%, but the measurement of efficiency is a very difficult problem in speed changes, because you are measuring a thing which is itself a very small fraction of the power you put in and the power you take out. When you try to measure that you sometimes find you have 105% efficiency and sometimes 91%, so that it is a very tricky business, this measuring efficiency at all.

Attempts have been made, taking the same housing, filling it with oil, putting electric heating elements in, measuring the electric energy necessary to maintain the housing at the same temperature above the surrounding atmosphere as obtains under a given load, and inferring that the energy which you dissipate in heat and lose in the speed changer is the same as in your subsequent experiment of heating up the oil inside, which sounds reasonable, but the scientific people say it is an unreliable way of measuring efficiency. We tried that very experiment that I have described and we got 98% or 98½%.

G. A. Kruse: Is the reducer reversible?

H. M. Edmunds: Yes. This is absolutely reversible, as you saw on that submarine steering gear, there are many cases in which we are asked to supply it for reversing.

There is another point which I would like to mention that that brings to mind, and that is that there is no lost motion, no backlash in this. I cannot move one of these even in the slightest degree without moving the other one. In any other form of speed changer with which I am familiar, except possibly a belt drive, that condition would not obtain. The teeth would have a little motion, from pressing on the tooth on one side to pressing on the other side. Sometimes that is quite an important point in haulage problems where you don't want to get any snatch. There is no snatch here at all and the reducer will run either way. But we would like to know which way it has got to run, because it is better to have the ring fall under load rather than lift. If the ring is to lift under load it is going to oppose gravity, and if we carry that experiment out we find the capacity or rating of a given speed changer should be 10% lower if the ring is to lift under load than if it is to fall.

J. R. Lewis: What would be the difference of efficiency between the two units, 300 r.p.m. motor, and the 1750 and the speed changer?

H. M. Edmunds: Well I don't honestly know what the efficiency of the H. H. motor is at 300 r.p.m. I would guess it would be about 80%, and the efficiency of the motor and speed reducer is 84.5%, but I couldn't find the efficiency of the slow speed D.C. machine. I dare say many people here could give us a good guess at the figure of the D. C. motor, 300 r.p.m. 15 h.p.

If you come to the alternating comparison with the induction motor, 300 r.p.m. the efficiency and the power factor—well the power factor is appalling in the squirrel cage induction motor, something like 50%.

G. H. Gildersleeve: If you have one of these installed for 8 to 1 ratio and a fellow wanted to change it to 4 to 1, could he change it by changing those rollers?

H. M. Edmunds: Yes, he can have a new pair of rollers to exchange for this 8 to 1; 4 to 1 would be rather drastic, but if you want to go to 7 to 1 without changing anything else, change this pair of rollers, slide along the bridge a little, he might have to file the holes a little so that the same end covers

could be used, roughly speaking it costs about a quarter of the price of the speed changer to change the ratio, that applies to either type.

J. L. Finnicum: As applied particularly to squirrel cage motors, a so-called across-the-line type, when you get into 45 h.p. are there exceptional stresses set up in the speed changer due to throwing a motor of that horse power across the line?

H. M. Edmunds: No, anything in the way of accelerated force that the motor can stand the speed changer can stand. In fact one of the features of the speed changer being combined with its motor, is that it is sort of a measure of protection for it, if the motor is going to blow its circuit breakers or burn out or fall out of step, when it is overloaded beyond a certain point, it acts in a manner of speaking, as a protection of the speed changer. Another thing about it, the action of the ring that I described and the means by which the pressure changes gives a sort of cushioning to shock loads, it prevents a sudden impact of the shock load. Driving a single cylinder compressor, if you start on the compression stroke and are up against a brick wall of compression, most gearing gives a great deal of difficulty and trouble under those conditions. We are rather fortunate in that respect because there is a cushioning effect due to the free ring moving on the other one, rather like a flexible coupling which has torsional flexibility.

A. Oslin: How about your vibration on higher speeds.

H. M. Edmunds: Well there is no vibration inherent in the speed changer itself. There is no vibration set up on the surfaces, they are plain cylindrical—if there is vibration in the motor it is going to vibrate the whole mechanism, if the motor has a magnetic vibration that is going to be there still, but there is no vibration inherent in this thing, and I believe that the C type of speed increaser is going to be a great improvement on the high frequency change of motor for the reason that it is one piece of homogeneous steel ground to a perfect cylinder throughout its length, therefore automatically balanced statically and dynamically, it won't be nearly as bad as if the motor shaft were itself doing the driving.

A. L. Reichert: There is no way of making that variable, is there?

H. M. Edmunds: Garrard has worked on that for a long time. This is a fixed ratio, whether that can be done, that is another story. There is a way by which you can have opposed cones and a ring between them and get variable gearing.

J. B. Mason: To come back to construction details again, what determines—I believe you spoke of that as a rolling up on the roll due to increasing load—what determines—what limits that?

H. M. Edmunds: The ring is going to be stretched more and more and increase compression which it exerts on being stretched.

A. Oslin: Those three rollers in alignment take up their greatest length, do they not?

H. M. Edmunds: Yes.

A. Oslin: Doesn't shorten the length?

H. M. Edmunds: No, the ring moves up on the outside of three rollers, the rollers being supported on the bridge construction.

S. L. Henderson: In case of that slow speed shaft or large shaft in the center, if that should be out of line with the smaller outside rollers, what would you expect then?

H. M. Edmunds: The three cylinders when they are pressed together must align automatically. These bolts placed under the bearing blocks are not tightened down until the ring is put over the three rollers and they are put in their correct position by the compression of the ring on the three of them, then the bolts are tightened down so they must be in line provided that the cylinders are.

A. F. Jones: In other words, in that type speed reducer you would have to have an anti-friction bearing.

H. M. Edmunds: You might have sleeve bearings.

A. F. Jones: Sleeve bearings will wear and probably throw one end up or one down, that would throw the rollers out of their alignment.

H. M. Edmunds: Not very much. Suppose it had a thousandth or two of freedom to wear up or down—with a fairly short length of the cylinder that wouldn't make bad contact, the rollers press into a flat of several thousandths of an inch width. I don't think you could get enough out of line due to wear in a bearing to upset the action, if that is what you mean.

A. F. Jones: Of course not while the machine is new, but later on when wear comes on you would have a lost motion in the bearing for the up and down motion, if it is in vertical postion.

H. M. Edmunds: But how much does a good model Ball or Roller bearing; how much looseness does it develop?

A. F. Jones: That is something else, but when you have a thing of that kind it wouldn't work very well for any length of time.

H. M. Edmunds: Yes it would be fairly long, Babbitt bearings have been used.

A. F. Jones: That would be compensated anyway, wouldn't it, by the floating ring under a load condition?

H. M. Edmunds: Of course it would impair the action very much if you didn't have one parallel to the other two because then it would weave—it would not be in line contact, it would have a contact, thin stripped from end to end if you put it at an angle, but no amount of wear that could occur in any bearing would be enough to produce that

A. L. Reichert: Would you recommend a ball bearing to a gear drive?

H. M. Edmunds: Yes.

I think ball bearings are all right for this; modern ball bearings are very good. In fact the art of making these speed changers has been built up on a ball and roller bearing practice, modern steels and modern grinding methods have made this speed changer possible.

S. L. Henderson: Do you have any applications where you still have a further reduction through gear drive?

H. M. Edmunds: Yes we have those too in a cement factory, Krupp in Germany have the same license that Crocker Wheeler Electric Mfg. Co. have in this country, they have used it a great deal in cement work, in these large kilns. They also make herringbone gears, so they are in position to know which they would rather build. They build these in preference to those.

S. L. Henderson: On the larger type running in oil, how high is the oil level in that?

H. M. Edmunds: It is enough at least so that the ring can dip into it, when the ring dips into it it makes an oil fog through the inside, as a rule it would be about the level of perhaps the top of this bridge. We also have a little oil pipe plug put into the housing to give you the height in filling it up you open that. Open the one on the top and fill until it flows out of the height level. I think that is better than having glass gauges, they break.

S. L. Henderson: The same thing holds true for the S type?

H. M. Edmunds: Yes.

F. C. Eckworth: How frequently would you suggest changing the oil?

H. M. Edmunds: It all depends, for speed increaser rather frequently, for speed reducer of the S type once a year would be enough.

Mr. Beck: Suppose one of the bearings of the motor that your speed changer is built into would wear considerably, would you have anything to change for it?

H. M. Edmunds: The motor bearings are all made for easy replacement, they are nearly all ball bearing now anyway.

J. R. Lewis: What would you say the life of the unit is?

H. M. Edmunds: That is a hard question to answer, they have been running for three or four years. The invention was only made in 1924, they are still running.

S. L. Henderson: Well from the observation that you have made in some of those running four years what would you estimate it to be?

H. M. Edmunds: I shouldn't like to be quoted as making a figure.

H. W. Eastwood: I would like to ask, first, Mr. Edmunds, have you a limitation on the maximum horsepower that you would tackle on this job?

H. M. Edmunds: Yes, we have built that biggest one, you remember the picture, C-47, that has a rating according to ratio and speed up to about 200 h.p.; about 50 h.p. 10 to 1 that is the biggest we have made yet.

An actual unit was built by the English Electric Company for 1000 h.p., 750 kilowatts for a turbine generator 6 to 1—that was the biggest.

H. W. Eastwood: I was thinking of that field for application.

M. W. Edmunds: That wants a lot of careful construction. If that reducer which the English Electric Company built many years ago were being built today it would be built differently. It did run, but the rings were not deep enough on this particular one I am referring to, they were too shallow and there was a good deal of creep; it wasn't built right. But today if we had occasion to build one we could design and build one and be quite sure it would be satisfactory for a load like that, it would have a number of these rolls in parallel, probably three or four of them, on long shafts, for greater capacity.

H. W. Eastwood: The other point was in connection with Mr. Stanton's question about the life of it; it seems to me that very likely that would depend a great deal on the service to which it would be subjected and also bearing on your statement during your talk that this really isn't any rare thing, I can see that there ought not to be any wear when the thing is running continuously, but in application such as is very common in the steel industry of course, where equipment is started and stopped with great frequency and this reducer is subjected to shocks with great frequency, it seems to me that every time you have a very sudden increase in torque there, every time you start from rest, there is some slipping between that slow speed roller and the middle roller, because of the distortion of the outer ring; it is almost infinitesimal, but if it is repeated often enough.

H. M. Edmunds: Perfectly true, the ring cannot move out of its position without the corresponding slippage between the two, it is a small matter, but it is there.

H. W. Eastwood: I wonder, if in view of that fact, you would recommend it for application for very frequent starting and stopping operations?

H. M. Edmunds: I don't think that actually there would be serious wear. It would be much more likely if there was a heavy thump coming at the same spot on a slow speed roller irrespective of what was hitting it. That is a very bad condition and under such conditions such as this all speed changer people, all gear makers, are very cautious.

As an example: For single cylinder compressors, they ask for heavy flywheels and for couplings with a good deal of tortional flexibility. Even in such cases we have sometimes arranged to move the slow speed roller around relatively to the coupling, and we ask the customer to do this, so as to relieve the particular spot which is getting the hammering.

As a further example: Suppose you put a ball race in a flying machine—common practice is to run it so many flying hours and change it for another one—this is because the bearings were intentionally loaded many hundred percent over the rated load.

We are conservative in rating so that we keep below the point where the surface of contact between the rollers does not deteriorate. If we are below a certain point, humanly speaking there will be no deterioration, while if we go above it the deterioration will be rapid.

Interesting in that connection many years ago before I went into this thing I had an interview with a famous engineer, Sir Charles Parsons, the inventor of the Parsons turbine. He is very much interested in this principle, which Mr. Garrard has incorporated into the design of the practical speed changer which we have here. Sir Charles said the thing you have to look out for is the "exfoilation" of the surfaces; he thought under the heavy pressure the steel would be apt to crack, and "exfoilation" was the term he coined for a "flaking off." As a matter of fact we haven't had any trouble of that nature.

The Deterioration and Maintenance of Oil

By COLBY W. BRYDEN*

The application of centrifugal force by the De Laval method to all types of problems involving the removal of solids from liquids and the separation of two liquids is so well known and has become so generally adopted that we are going to discuss oil maintenance from a somewhat different angle and talk about what actually happens to the oil in service. Before going on, we would like to say a word of appreciation for the hearty co-operation that the larger and more progressive oil companies are giving to manufacturers of purifying equipment as well as to their own customers by recommending adequate means of keeping the oil in proper condition to give efficient lubrication over long periods of time. These oil companies realize that they are selling a service rather than a commodity and the results that the customers obtain from their products depend on the length of time that the oil will give a satisfactory degree of lubrication and fulfill its purpose.

The process of deterioration in oils are more or less common to the various services on which they are employed. For this reason we will first take up transformer oil and see what actually happens to the oil in service and what are the causes of the deterioration. We can then follow these same reactions into other types of oils used in lubrication work.

Transformer Oil

The main functions of transformer oils are—

1. as a cooling medium to conduct away heat, and
2. as an insulating medium.

The two things which transformer oil must do, and do well, are outnumbered by the things that the transformer oil must not do, which are as follows:

They must not thicken in service, thus retarding the flow and causing excessive heating.

They must not oxidize readily to form acid products which attack the insulation as well as attacking the copper and forming metallic soaps.

They must not form sludge which collects on the windings or on the cooling coils thus reducing the heat transferred and causing local hot spots or general over-heating.

Mineral oils are generally used for this service and it is possible by distilling separate desired fractions and refining to obtain a product which holds its qualities during long service. It is interesting to note that oils which are highly refined or perhaps over-refined and some white oils do not sludge for a long period but will develop acidic products that attack the insulation, while non-acid forming oils sometimes give heavy sludge. It is necessary to strike a happy medium and it is impossible to base specifications on reaction products as a criterion.

*The De Laval Separator Company, New York.

Air Absorption

Oils which are exposed to the atmosphere can dissolve from 10 to 20% by volume of air. This absorption of air can take place almost instantaneously. This air in solution furnishes a considerable amount of the oxygen which is required in the oxidation of the oil as it occurs in service. As the oxygen content of the oil is used up, more air and hence oxygen is taken into solution so that the reaction is continuous.

Oxidation

The oil in a transformer contains air and possibly some water, and it is subject to high temperature, thus giving three conditions necessary for deterioration. Some of the oil is oxidized giving reaction products which can be water, oil soluble and oil insoluble, acids which may undergo a further change to form neutral compounds, sludges and metallic oxides. Some of the acid products attack the insulation doing considerable damage. The sludges may collect either on the windings or on the cooling coils, thus impairing heat transfer, and causing hot spots and general over-heating. The oxidation process is materially increased by contact with some metals which act catalytically, and among the more important of these are copper and copper alloys such as brass. The sludge itself is a catalyzing agent which assists in the formation of more sludge.

Conservation

A number of transformer manufacturers are trying to prevent this oxidation by the use of conservator type transformers or by other means such as the Westinghouse inert air principle. Thus it is understood that the oil as put in the transformer contains a considerable amount of air from which the oxygen will be used up during the first few months of service thus forming sludge and other products of deterioration. Then if no more air is allowed to enter the oil this deterioration process should cease. It is frequently considered advisable to purify the oil in transformers protected by conservator or inert air devices after a few months of service, thus removing the sludge and any of the deterioration products which would in themselves aid further deterioration. This purification must be done without exposing the oil to contact with air since even momentary exposure will allow the oil to become saturated with air. This purification and dehydration is frequently accomplished by means of purifying centrifugally with conservator attachments, and if it is accomplished, the transformer can remain in service for a long period without any further attention to the circulating oil.

Water Solubility

Many operators of transformers have wondered why the oil, while showing a high dielectric strength after being purified centrifugally at around 50° C. would show a marked decrease in dielectric strength after several days of standing in storage tanks or transformers at lower temperature. This loss in dielectric strength is due to the ability of oils to actually take water into solution, and this water in solution cannot be removed commercially by any present method. The ability of the oil to take water into solution increases with rises in temperature, and oil from which all free water has been removed at high temperature can precipitate out a considerable amount of dissolved water when the temperature drops. This precipitated water is in very fine particles thoroughly dispersed throughout the oil, and can easily cause break-down. It is now a known fact that oils can actually take into solution from 300 to 700 parts per gallon of water and that is the reason that we are recommending the purification of transformer oils at low temperature.

Note: The research work on water solubility of oils was performed by Mr. M. A. Dietrich under the direction of Dr. A. E. Flowers in the Research Laboratories of The De Laval Separator Company.

The collection of water at the bottom of the transformer is often surprising and can be accounted for in part by water in the atmosphere condensing on the top of the tank due to temperature changes and dropping into the oil, then due to its heavier weight gradually sinking to the bottom. The water that gets into the oil in this fashion is not particularly dangerous. It is the water that enters the oil in the air and in the form of water vapor as well as the water which is formed as a product due to the oxidation of the oil that becomes thoroughly dispersed throughout the body of the oil and is most conducive to break-down. When we visualize this continual process, it is easy to see how collections of several gallons of water in the base of the transformer can occur. It is much easier for oil to absorb water in the form of water vapor than in the form of free water as it occurs in drops and the humidity of the atmosphere has quite a bit to do with the moisture content of oil.

A number of tests have been made to determine how transformer oil will perform in service and these tests include the Snyder life test and many other methods which have been used abroad whereby oil was heated in contact with air or oxygen for a number of given hours possibly in the presence of catalyzing agents such as metals prepared in various ways and then the sludge formation is noted and some attempt is made to judge the life and performance of the oil. In the case of the Snyder test 12 to 14 days is considered satisfactory and 20 days is good if no sludge appears by that time.

Transformer oil can be brought back to its dielectric strength by means of centrifugal purification, thus removing sludge and water but ultimately the demulcibility lowers and the acidity increases so that in some instances it is necessary to adopt chemical treatments such as the use of tri-sodium phosphate and certain earths in order to bring back demulcibility and reduce acidity. It has been found however, that chemical treatments cannot bring back the life test even where the acid process is used. For general purposes the chemical treatment need only be used after very long periods of service.

Circuit Breaker Oils

The oil used in circuit breakers collects contamination from different sources than that in transformers. It may collect water more readily, and in the operation of the breaker it picks up considerable quantities of carbon which must be removed in order to keep the oil in satisfactory condition. The oil in the circuit breakers is not usually heated, and oxidation as well as acid products and sludge do not bother it as they do in a transformer. Some studies have been made to determine the effect of fine particles of copper which are dispersed through the oil when a luminous arc occurs, but these need not be discussed at present.

Circuit breaker oil can be easily maintained in proper condition by centrifugally purifying to remove water and heavy carbon, then passing the oil through a filter press to remove the colloidal carbon. Such treatment is all that is required in most cases and it is easier to keep the oil in good condition if it is cleaned at frequent intervals rather than allow the carbon to accumulate into large quantities. When circuit breaker oils require chemical treatment on account of demulcibility or acidity the same process can be applied that is used on transformer oils.

Turbine Oil

Turbine oil is subject to many of the same conditions that effect transformer oil. The oil in the bearings is frequently at much higher temperature than the main body of the oil due to the shearing action in the maintenance of the film. This hot oil which contains both oxygen and water in solution is sprayed from the bearings in fine particles, thus giving the ideal conditions for oxidation and deterioration. The oxidation usually takes the form of acid products which later combined with more water and by further oxidation form various types of sludges. The reaction products such as the acids and sludges have varying solubilities in water and in oil depending on the temperature. As the sludge which is soluble in the oil at high temperatures condenses on the coolers the temperature of the oil rises, the sludge acts as a catelizer which aids further oxidation, and we have practically the viscous circle with each element furthering the deterioration.

In the past many turbine operators have kept the oil in fair condition by purifying it centrifugally at fairly frequent intervals, thus removing sludge and water. While others have taken the further precaution to purify the oil continuously during the operation of the turbine, thus preventing any collection of water in the oil and considerably retarding the oxidation as well as removing insoluble sludge as fast as it is formed. Neither of these methods is entirely successful in preventing the formation of sludge on the coolers, as this sludge is apt to be insoluble at high temperatures though it easily condenses on the cooler tubes.

Recently, or rather during the last three years, we have tried to prevent the formation of sludge by continuously removing as much as possible of the intermediate deterioration products such as the sludge forming acids. In several instances the oil

as it is going from the turbine base for purification is washed with condensate which removes from the oil certain acids having a greater solubility in water than in oil. By means of this continuous water wash and purification it has been possible to keep turbine oil in service over a period of three years with a very slight increase in acidity as determined in milligrams of K O H and practically all formation of sludge on the coolers has been eliminated, while the formation of sludge deposits in the base of the turbine has also been greatly reduced. This system has changed the character of the sludge which is removed from the bowl of the De Laval Purifier and much less sludge was taken out during the entire three years run than would normally be removed by the ordinary methods of purifying the turbine oil. This shows that the attempts to prevent the formation of sludge have been in large measures successful, and the oil was still in excellent condition after a long period of use.

With this system it is necessary to run continuously as the relative solubility of the oxidized products in water and in oil changes with age, and these products must be removed when they are first formed. From the experience already gained it is evident that the most efficient purifying system of turbine oil is also the most economical both from the standpoint of oil consumption and maintenance.

DIESEL ENGINE OILS

Lubricating Oil

Diesel engine lubricating oil is subject to the same types of contamination which are found in lubricating systems of steel mills, drawing machines and other places where oil is used as a lubricant in the steel mill. In a Diesel engine it is a question of outside contamination such as water, carbon, sand, dirt and metal particles. The sludging or oxidation does not bother us greatly except as it is manifested in the formation of carbon. The carbon may come from contact of the oil with hot piston heads or from incomplete combustion of the fuel which products work down into the crankcase. Here again it is a question of continuously removing the contamination which may be abrasive faster than it is built up in the normal operation of the engine. The colloidal carbon is not harmful as it is more in the nature of a lubricant than an abrasive, but the heavier forms of carbon can cause damage and they, as well as the other type of contamination should be removed.

Some tests were recently made by a Diesel engine builder on three engines; one operating without any purifying system, one operating with a common filtering system, and the third with centrifugal purification. After a run of a few thousand hours the engines were taken down and measured for wear in the bearings. The difference in the condition of the three engines was surprising and again the most thorough system proved to be the most economical.

Diesel engine lubricating oil is consumed fairly rapidly which necessitates the adding of make up oil. This continual sweetening of the oil has been thought to keep the main body of the lubricant in fair condition, but the reverse is actually true. The contamination and the solid particles keep building up regardless of the amount of make up oil that is used, and unless they are removed, poor lubrication with the attendant troubles and expenses will be the result.

Diesel Fuel Oil

Many Diesel fuel oils are bought on a gravity basis and therefore are not clean, as it is often the practice to mix heavy residual oil with distillate to obtain a certain gravity. Fuel oils often contain considerable amounts of water, sulphur, carbon, dirt and tarry sludge. If this dirty oil is fed to the engines the results are undue wear of the fuel pumps and injection valves, excessive carbonization, poor and incomplete combustion and the formation of acid products which may cause pitting or corrosion of the engine parts as well as contaminating the lubrication which will cause bearing trouble. The results of purifying fuel oil are well known, and several engine operators consider that clean fuel oil is more important than clean lubricating oil.

In cleaning fuel oil temperature often plays an important part in reducing viscosity and density to a point where it can be satisfactorily separated from the water which is often bound up in the oil. This is particularly true in the case of the heavier oils where recommendations for handling them must be left to engineers who have had experience in that work.

GENERAL STEEL MILL LUBRICATION

Oils employed in steel mills are probably subjected to greater contamination than any other service. While lubricating efficiency is of as great importance as in central stations since 24-hour service is required in both places. Usually steel mill operators are willing to buy good grades of oil and plenty of it, but in the past some mills have made considerable savings by taking better care of their oil after it was placed in service thus obtaining longer life, better lubrication and lower maintenance.

Lubrication of the Mill

While some of the older mills are still using individual lubricators for the various parts, the more modern plants have continuous circulating systems for lubrication with several thousand gallons of oil in each system. The usual contamination in this work is water, scale, dust, metal particles, carbon and other impurities, some of which come from the dirt laden air. The problem is similar to that which exists in the case of the Diesel engine lubricating oil, namely, to remove these impurities faster than they build up. This can be done by continuously purifying from 10 to 20% hourly of the entire amount of oil in the system. Due to the danger of emulsions it is necessary to keep the oil practically free from water, and it is a simple matter at the same time to efficiently remove the dirt and solid matter. The general precaution, that of duplicate equipment, should be followed in laying out the circulating system for the mill.

Drawing Oil

Drawing machines and straightening machines seem to have been discriminated against in the laying out of systems for lubricating the work, the dies and the rolls. In many cases small storage tanks of oil are used with the circulation of the oil at high

speed with practically no chance for the contamination to settle out or no provision for proper purification. In the drawing machines the oil is soon contaminated with dirt, scale and lime, which quickly thickens it so that it does not penetrate to the point where the cutting is to be done, and the entire tank full of oil must be discarded. Additional bad features are the wear on the dies and the poor finish on the work when the lubrication is not efficient. At one plant this condition has been greatly improved by the use of a larger amount of oil in the system, a higher rate of circulation and efficient centrifugal purification of the oil with the result that a large saving has been made in the oil alone. The lime consumption has been decreased by about 50% and much better work is being turned out with correspondingly greater life of the dies.

The operation of the straightening machines can also be greatly improved by supplying more efficient lubrication which will result in less wear on the rolls, better finish on the work and some oil economy although the initial outlay for purifying equipment and an adequate circulating system may be greater than in the case of the older methods.

Rolling Mill Engines

Rolling mill engines usually receive a great deal of abuse and they are expected to perform continuously, as a breakdown would be expensive. We now have a pretty complete idea of the various things that happen to the oil, particularly in steel mill work, and the rolling mill engine lubrication oil is subject to the usual ailments. It is interesting to see the difference in operation when proper care has been taken to afford efficient lubrication to the rolling mill engines. The proof that these engines appreciate good lubrication is shown by a number of reports which mention lower oil consumption, keying up the bearings every eight weeks instead of every week and less men required on maintenance work.

Generally speaking, the life of the oils used in steel mills can be materially lengthened by proper care, and at the same time large economies can be effected from the standpoint of maintenance and operation. The service is harder than that which is met in most factories or power plants due to the additional sources of contamination, but there is nothing mysterious about the deterioration of the oil. A proper understanding of the reasons for the dropping off in lubricating efficiency enables the engineer or superintendent responsible for that part of the operation to take the necessary steps to afford better lubrication, and it will be found that the initial cost of the equipment will be small when compared with the ultimate savings.

It has been proven that oils in general do not undergo changes which kill their lubricating value. They merely pick up outside contamination and certain parts of the oil may be oxidized to form deterioration products which are harmful but these products and contamination can be removed from the oil and thus return it to its initial value as a lubricant.

OBITUARIES

Abraham F. Huston, 78, since 1925 chairman of the board of the Lukens Steel Co., Coatesville, Pa., died at his home there January 12. He was born in Coatesville July 7, 1852, the son of Dr. Charles Huston and the grandson of Dr. Robert Mendenhall Huston, one of the faculty of Jefferson Medical College, Philadelphia. A. F. Huston was graduated from Haverford College and in 1872 entered the plant of the Lukens Rolling Mills, founded by his grandfather, the predecessor of the Lukens Steel Co. He progressed through various departments, both plant and office, gaining a complete knowledge of the business. Prior to 1925 he was president of the company for many years, succeeding his father in that office. He was president of the Coatesville Trust Co., director of the Commercial Trust Co., and member of the Manufacturers' Club, Philadelphia. In 1902 he was president of the American Steel Manufacturers' Association and at one time was a committee member of the American Iron and Steel Institute.

Frank J. Brittingham, age 46 years, an associate member of the Association of Iron & Steel Electrical Engineers, died at his home, 2118 Beechwood boulevard, Monday, January 27.

Mr. Brittingham represented the Cleveland Crane & Engineering Company, being District Manager at Pittsburgh.

He was born in Michigan City, Ind., May 9, 1884. After his graduation from Purdue University in 1905 he served an apprenticeship course with the Westinghouse Electric & Manufacturing Company. In 1911 he became associated with the Cleveland Crane & Engineering Company, and later was appointed District Manager with headquarters at Pittsburgh. Mr. Brittingham is survived by his widow and one son.

John H. Lendi, a member of A. I. & S. E. E., died January 18 at his home, 5133 Sheridan road, Chicago, following a short attack of angina pectoris. Mr. Lendi had been in failing health for over a year.

Mr. Lendi was born in Dubuque, Iowa, June 25, 1874, from which place the family moved to Terre Haute, Ind., while he was still a boy. He graduated in electrical engineering at Rose Polytechnic Institute. He became electrical engineer of the Universal Portland Cement Company in 1910 and continued in that capacity until the time of his death. Mr. Lendi is survived by his widow and one daughter residing in Chicago; also by his mother and two sisters who reside in the old home at Terre Haute, Ind., from which the funeral was held Monday, January 20.

E. H. Pearson, New York District Manager of Electric Machinery Manufacturing Company, succumbed very suddenly on the morning of December 7. His death is a great loss, not only to his associates in the Electric Machinery Manufacturing Company, but also to his friends throughout the country.

Items of Interest

PERSONNEL CHANGES

George Gordon Crawford was elected President of the Jones & Laughlin Steel Corporation at a meeting of the directors of the corporation January 28. He was also made a member of the board of directors and of the executive committee.

For about 23 years he has been President of the Tennessee Coal, Iron & Railroad Co., Birmingham, a subsidiary of the United States Steel Corporation. It is understood that he will assume his new duties in the immediate future.

Mr. Crawford is a native of Georgia and was graduated from the Georgia School of Technology in 1890 and then studied for two years at Karl-Eberhard University at Tubingen, Germany. He was assistant superintendent of the Edgar Thomson blast furnaces of the Carnegie Steel Co. from 1895 to 1897, superintendent of the blast furnace and steel works of the National Tube Co. at McKeesport for the next two years, and then manager of the National Tube Co., at McKeesport, Pa. In 1907 he was elected President of the Tennessee Coal, Iron & Railroad Co. with offices at Birmingham, Alabama.

Herbert C. Ryding was appointed President of the Tennessee Coal, Iron & Railroad Co. at Birmingham on January 28, succeeding George Gordon Crawford, who resigned, effective January 31. Mr. Ryding has been vice president in charge of operations of the Tennessee company since 1917, and has had many years of service with the corporation.

At a recent directors' meeting of the Union Carbide Company and the Electro Metallurgical Company, both units of Union Carbide and Carbon Corporation, Benjamin O'Shea, formerly President, was elected chairman of the board of each company. Fred H. Haggerson, former Vice President of the two companies was elected to succeed Mr. O'Shea as President; and F. P. Gormely was elected Vice President and General Manager of both companies.

E. A. Leinroth has been appointed Assistant Sales Manager of Electric Service Supplies Company, Philadelphia, Pa., succeeding I. W. Schmidt, who has been promoted to Western Sales Manager, with headquarters in Chicago, Ill.

John James Noell, District Engineer, of the Holophane Company, Inc., who has had headquarters at the Fairyland Club, Lookout Mountain, Tenn., is now located in Birmingham, with headquaters at 2501 Montevallo road, Mountain Brook.

Mr. Noell spent several years in the engineering organizations of the DuPont Company and Stone & Webster, Inc., Boston. He takes with him a wide experience in electrical work and is rapidly removing dark spots from his native sunny southland.

Paul H. Butler, who for five years has been engaged in sales and engineering work out of the New York office of the Delta Star Electric Company, Chicago, dealing with utilities and industrials, has been appointed Manager of the New York office.

From 1921 to 1924 he was a member of the firm of Harris and Butler, representatives of several electrical manufacturers at Philadelphia.

The preceding six years Mr. Butler was Manager of the New York office of the Pittsburgh Transformer Company which company he also served from 1908 to 1913.

Mr. Butler is a member of the Engineers Club, New York.

H. O. Anderson, formerly a Special Sales Engineer for the Rockbestos Products Corporation, New Haven, Conn., has been appointed Sales Manager of the company, which manufactures a complete line of asbestos insulated wires and cables.

At the annual meeting of the stockholders of Stimple and Ward Company, George E. Stimple, formerly Secretary, was elected President and Secretary. The A. I. & S. E. E. of which he is a member wishes to extend their sincerest congratulations.

J. B. Crane has been transferred from the New York office of Combustion Engineering Corporation, and has been appointed their Pittsburgh District Manager with offices at 1606 First National Bank Building, Pittsburgh, Pa.

The Pittsburgh office of The Reliance Electric & Engineering Company has moved to new and enlarged quarters at 2300 Koppers Building in Pittsburgh.

B. J. Ballard continues as District Sales Manager with C. V. Gregory as assistant.

The Pyle-National Company's Boston office formerly located in the Little Building has been changed to Room 901 of the Statler Building. Carl S. Geis is in charge.

The Rockbestos Products Corporation of New Haven, Conn., have opened a district sales office at 2143 Railway Exchange Building, St. Louis, Mo. F. W. Allen, who has been connected with the Chicago sales office of this company, will be in charge.

E. H. Clark has joined the selling organization of the Rockbestos Products Corporation. He will be connected with the Chicago district office. Mr. Clark has been in the electrical field for the past sixteen years as General Superintendent of the Mayville plant of the Youngstown Sheet & Tube Co.; in construction work with the Wisconsin Gas & Electric Co. at Kenosha, Wis., and as salesman for the Electric Appliance Co. of Chicago.

William T. Martersteck sailed February 8 on the Aquitania enroute to Leningrad, U. S. S. R. He will form a part of the group of Freyn engineers who are in Leningrad in connection with a contract between the Freyn Engineering Company and the Union of Socialist Soviet Republics for technical assistance in the latter's iron and steel industry.

Mr. A. W. Mohrman, formerly Electric Sup't., Follansbee Bros. Co., Follansbee, W. Va., has been appointed Ass't. Chief Electrician, Newton Steel Co., located at Monroe, Mich.

W. S. Orr and Vance A. Cronk sailed February 1 on the Bremen enroute to Leningrad, U. S. S. R. They returned to the United States a month ago in connection with engineering services which the Freyn Engineering Company is rendering to the Soviet Government.

The board of directors of the American Rolling Mill Company on Tuesday, January 7, voted to enlarge the executive staff of the corporation by creating the offices of Chairman of the Board and Vice Chairman, in addition to the existing positions.

George M. Verity, President of the company since its inception, was elected to the position of Chairman, and Joseph H. Frantz, First Vice President for the past 16 years, was named Vice Chairman.

To succeed Mr. Verity as President the board elected Charles R. Hook, who has been Vice President and General Manager for the past seven years. Mr. Hook as President will retain the duties of General Manager.

In addition, Calvin Verity, who has been Treasurer and Assistant General Manager, was named Vice President and Assistant General Manager.

S. R. Rectanus, assistant to Calvin Verity, was made Vice President in charge of operations.

G. W. Davis, Assistant Treasurer, was elected Treasurer, a position held until recently by Calvin Verity.

E. N. Millan, Chief of Construction, was made Chief Engineer of the company.

The Roller-Smith Company, 233 Broadway, New York, N. Y., makes the following announcements of changes in sales organization:

M. W. Seymour is now associated with the New York office as a Sales Engineer. Mr. Seymour is a graduate of Brown University and for several months prior to his connection with the New York office he was located at the company's works in Bethlehem, Pa.

H. D. Stier, 101 Marietta street, Atlanta, Ga., now represents the Roller-Smith Company in the states of Alabama, Florida, Georgia, North Carolina and South Carolina.

H. N. Muller Company, First National Bank Building, Pittsburgh, Pa., now represents the Roller-Smith Company in Western Pennsylvania, West Virginia and the Youngstown district in Ohio. Associated with H. N. Muller are H. E. Ransford and F. E. Harper.

The Roller-Smith Company has recently issued two new bulletins as follows:

Bulletin No. 300 covering various types of instruments for resistance measurements, namely, Types COM, GOM and SOM Ohmmeters and Type HTD Circuit Tester.

Bulletin No. 210 covering Portable Direct Current Ammeters, Milli-ammeters, Voltmeters, Millivoltmeters, Volt-ammeters, Galvanometers, Circuit Testers, Shunts, Multipliers, etc.

James P. Dovel has severed his connection as Vice President and Manager of Furnaces of the Sloss-Sheffield Steel & Iron Company and will devote his entire attention to Blast Furnace Engineering and Contracting business.

The Clark Controller Company of Cleveland, Ohio, announce the appointment of H. G. Mouat, 1116 Martin Building, Birmingham, Alabama, as their Southern District Representative.

DESIGNERS

Thoroughly experienced General Machine Designers, capable to design heavy machinery built of Cast Iron, Cast Steel or Welded Rolled Steel Shapes and Plates. Large firm, Eastern United States. Give education, experience, age, and where employed. Your reply will be held confidential. Write: Designer, care of Iron & Steel Engineer, Empire Building, Pittsburgh, Pa.

NEWS FROM THE MANUFACTURERS

SOLVING POWER FACTOR PROBLEMS WITH SCALE

The General Electric Company announces a new system of solving power factor problems. This is a graphic method involving simple computations only, and is described fully in a new publication, GET-191, entitled "Solving Power Factor Problems by Scale."

Heretofore the calculations relating to power factor and its improvement have been difficult for those most vitally interested, and at best did not visualize the conditions that might result from different treatments. In brief, the new system consists of laying down to scale the known factors and then scaling the unknowns. Many problems are illustrated in the booklet, as well as several solutions for one problem, so that there should be no difficulty in applying the simple rules. Thus the effects and probable costs of improving power factor by unity-power-factor synchronous motors, by capacitors or by synchronous condensers may be scanned and compared. Increased load-carrying capacity is readily measured and rearrangements suggest themselves.

All that is needed for computation is a scale, a pencil and paper. A right-angle triangle with the sides in proportion 3 to 4 to 5 is furnished with the booklet and this, together with a small T-square, save time in the calculations and make the work more accurate.

In the actual calculations, horsepower at unity-power-factor is expressed in terms of inches measured horizontally. Lagging power factor is represented by lines slanting upward to the right, and leading power factor, by lines slanting downward to the right. Where it is desired to improved lagging power factor, and to determine the amount of leading power factor necessary to accomplish this result, a triangle is laid out using the horizontal unity-power-factor line and the upward lagging power factor line. Then by applying the 3-4-5 triangle the necessary corrective value may be established. Other problems are solved in a similar manner.

ELECTRICALLY WELDED BASES

The Reliance Electric & Engineering Co., Ivanhoe Road, Cleveland, Ohio, have adopted bases constructed from heavy steel plate, electrically welded for motor-generator sets which they manufacture in sizes up to 250 Kw.

The tops and ends of the welded base are completely closed so that no dirt can collect underneath.

The pads upon which the motor and the generator rest are lined up horizontally and vertically by using the machines themselves as patterns for doing the aligning. Such shims as are needed are placed between the pads and the base. The pads with the shims beneath are then welded to the base. Horizontal alignment is maintained by the use of tight-fitting dowels. These are driven into the pads, through two diagonally opposite feet of each unit. One or more units may be removed from the base without worry about the alignment when they are to be replaced.

E-M IMPROVED CELL TYPE CAPACITORS

The Electric Machinery Manufacturing Company of Minneapolis, Minn., has recently announced a new improved line of cell type Capacitors for use in power factor improvement. These capacitors are built for either indoor or outdoor use, the outdoor type capacitor being completely enclosed in a sheet steel housing.

The E-M capacitor consists of an assembly of capacitor cells securely mounted on a rugged steel stand. Each cell is a complete capacitor unit, and by connecting two or more cells in parallel, any desired kv-a. may be obtained.

A feature of the E-M cell type capacitor is the cylindrical roll, which is the "working element" in the capacitor cell. The cylindrical form of construction prevents undue mechanical stress on the tissue, thus insuring long life.

The individual capacitor rolls are mounted on a supporting rack and sealed under oil in the cell. The cell containers are substantially built from 18 gauge sheet steel and have welded oil tight joints. Stands for holding the cells are of welded sheet steel construction.

AIR COOLED AIR COMPRESSORS

Ingersoll-Rand Company announces a new line of air-cooled, two-stage air compressors, known as the Type 30. V-type belt drive is employed. Both motor and compressor have ball bearing.

The units are self-contained, the motor and compressor being mounted on a steel base, which is attached to the top of the air receiver. The latter, which is made of heavy pressed steel, is built to withstand a working pressure of 200 pounds. With this arrangement, no special foundation is required for correct alignment of the compressor and motor. The compressor is ready to operate as soon as the electrical connections to the motor have been made and the crankcase filled with oil.

The intercooler is located behind the fan-type flywheel, and a constant current of circulating air is driven directly across the cooling coils. This reduces the temperature of the discharge air.

Automatic start and stop control, furnished as standard equipment, operates independently, but in conjunction with the unloader. When the pressure in the air receiver reaches a point at which the regulator is set to unload, the motor is automatically shut off. A centrifugal governor allows the air in the high-pressure cylinder and intercooler to exhaust through the crankcase. This prevents the compressor from starting against a load.

Honed cylinders and two oil control rings reduce the oil in discharge air to a minimum. Each piston is run into its respective cylinder, insuring a perfect oil seal.

The base of the compressor unit forms a reservoir for the oil, and no oil pump is required. A bayonet gauge gives positive indication of the amount of oil in the reservoir.

A self-cleaning air cleaner keeps dirt out of the compressor and requires no attention. The compressor is entirely enclosed, and no dirt can get into it to wear out the working parts.

A balanced crankshaft, eliminates destructive vibration.

This improved two-stage design reduces power from 10 to 30 per cent. At the same time, less floor space is required.

It is built in four sizes: ¾, 1½, 3, and 5 horsepower. All sizes are built for a working pressure up to 200 pounds continuous duty.

A free copy of bulletin 3060 can be obtained by addressing the Ingersoll-Rand Company, 11 Broadway, New York.

NEW TRADE LITERATURE

The Westinghouse Electric and Manufacturing Company announces the release of a new four-page descriptive leaflet on Jet Condensers.

The leaflet describes in detail the construction and application of jet condensers. Many illustrations are used; one of which shows a typical installation layout of a jet condenser serving a turbine-generator unit.

This publication may be secured by sending a request for Descriptive Leaflet 20435 to the Westinghouse Electric and Manufacturing Company.

The "1930" series of AutoStart Grinders is the subject of an interesting bulletin recently issued by U. S. Electrical Manufacturing Company, with head offices in Los Angeles. The bulletin describes the numerous new features embodied in AutoStart Grinders, including the new chip shield which is said to dispense with the need for bothersome goggles; the improved method of permitting proper guard clearance so that the tool holder can be maintained at a uniform distance from the wheel as the wheel grinds down to a smaller diameter, and a third feature that is interestingly described is the method of locking the tool rest. Sent on request.

The Martindale Electric Company are distributing a bulletin on their Lathe Type Undercutter. Copy may be obtained by writing to their Cleveland address.

Colt's Patent Fire Arms Manufacturing Company have gotten out an announcement on their new QuadBreak Switch. The announcement gives a very complete description of this new line of switches. Copy may be obtained on request.

The Bristol Company of Waterbury, Conn., have

prepared a very attractive 30-page booklet on "Recording Pressure and Vacuum Gauges." List of charts available for Bristol Recording Pressure and Vacuum Gauges are also included. Interested parties may obtain copies on request.

An attractive bulletin, "Modern Motive Power for Switching Service" is now being distributed by the George D. Whitcomb Company of Rochelle, Ill. Copy of the bulletin may be had by writing the above concern.

Bulletins on "Plugs and Receptacles for High Frequency Tools" and "Type FSQ Interlocking Safety Switch Condulet," have been prepared by the Crouse Hinds Company of Syracuse. Interested parties may obtain copies on request.

The Weston Electrical Instrument Company of Newark, N. J., have developed a new instrument, known as Model 553 Heat Flow Meter. It is said it will measure heat flow with almost the same ease and facility as the measurement of current with an ammeter. Further information may be had on the instrument by writing the above company for Bulletin 4800.

The Reliance Electric & Engineering Company are distributing a new 24-page illustrated bulletin No. 205, covering their Type T Heavy Duty Reliance Planer motors for Reversing Service.

The Corrigan, McKinney Steel Company have just started up a battery of refractory recuperative pits at their plant at Cleveland, Ohio. This installation consists of seven holes—each hole with capacity of eight or ten ingots, depending on ingot size. By-product coke oven gas is used for fuel in these pits.

The Bourne-Fuller Company, of Cleveland, have just placed in operation a 5-hole pit of the same design, except for the use of a Blaw-Knox Preheater in place of refractory recuperators. These pits are fired with producer gas.

The Otis Steel Company, of the same city, is starting construction of a battery of the same design of pits at their Riverside Works.

The Chapman-Stein Company at Mount Vernon, Ohio, is designing and constructing these pits.

Recent Lifting Magnet sales by The Ohio Electric & Controller Co. to iron and steel industry included: Three 55" and two 65" Magnets to Great Lakes Steel Corp.; four 36" Magnets to Chile Exploration Co. to be mounted on the booms of a steamer which they will use for collecting scrap from along the coast of Chile for use in copper extractors; four 65" Magnets to the General Steel Castings Co., Eddystone, Pa.; one 55" Magnet to Southern Malleable Iron Co., East St. Louis; one 55" Magnet to Carnegie Steel Co., Clairton, Pa.; one 36" Magnet to Universal Exploration Co., Jefferson City, Tenn.

National Tube Company has purchased three Freyn-Design Continuous Stock-line Recorders, two of which are for installation at its National Works, while the third is for its Lorain Works. Carnegie Steel Company has purchased a second Freyn-Design Continuous Stock-line Recorder for its Edgar Thomson Works. Pittsburgh Crucible Steel Company has also purchased one of these equipments for use at its Midland Furnaces.

A REVIEW OF METHODS OF THE CHEMISTS OF THE UNITED STATES STEEL CORP. FOR THE SAMPLING AND ANALYSIS OF COAL, COKE AND BY-PRODUCTS

By W. A. SELVIG†

This book, prepared by a committee of chief chemists from the coal and coke laboratories of the constituent companies of the United States Steel Corporation, is a complete revision, with many additional methods, of the former edition which was published in 1923. It is a manual of 334 pages.

Part 1 includes sampling of coal, sampling of coke, laboratory preparation of samples, physical tests for coal and coke, and methods of analysis. The physical tests include methods for screen and sieve tests, weight per cubic foot of crushed bituminous coal and of coke, fusion point of coal, float-and-sink test for coal, and shatter test, tumbler test, apparent and true specific gravity of coke.

The methods of analysis give detailed instructions for the proximate and ultimate analysis of coal and coke, determination of forms of sulphur, determination of phosphorus and metallic iron in coal and coke. A complete scheme for analysis of coal and coke ash is given, also methods for fusibility of coal ash, ultimate analysis of pitch and tar, calorific value of solid and liquid fuels, and a method for the determination of the relative yields of coke and by-products from bituminous coal by the progressive distillation method.

Part 2 includes comprehensive methods for the examination of products of and materials for the ammonia recovery plant and gives methods for the examination of ammonia liquors and still waste, determination of cyanides and phenols in waste and natural waters, examination of sulphuric acid, saturator liquor, lime, ammonium sulphate and ammonium chloride. Detailed methods are given for the examination of light oil and benzol products.

Part 3 gives methods of sampling and analysis of coal tar and tar refinery products such as tar and crude tar products, pitches, creosote oils, crude tar acids, and crude and refined napthalene.

The book has five useful tables on constants for sulphuric acid solutions, temperature corrections for specific gravity of sulphuric acid, conversion factors for gravimetric and volumetric analysis, specific gravity of ammonia solutions, temperature corrections for specific gravity of ammonia solution, and a list of the chemical elements and atomic weights. It contains 54 illustrations giving construction details of apparatus, and complete directions are included for the preparation and standardization of standard solutions.

The methods of analysis and testing have been carefully tested and standardized in the laboratories of the constituent companies of the U. S. Steel Corporation and therefore represent the standard practice of these laboratories. The chemists of the committee responsible for the preparation of the book have made readily available a large amount of valuable information on the analysis and testing of coal, coke, and by-products. The book is a valuable contribution to the literature on methods for the analysis and testing of coal, coke and by-products.

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†Chief Chemist, Bureau of Mines Experiment Station, Pittsburgh, Pa.

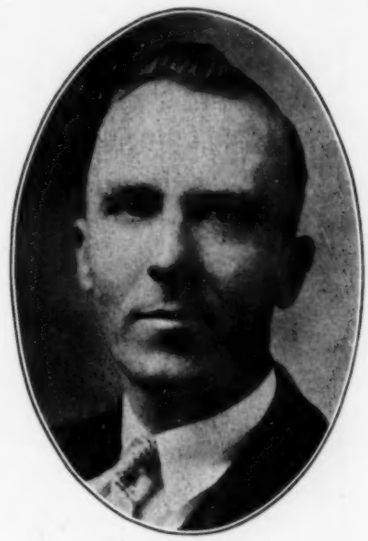
BIRMINGHAM DISTRICT SECTION

EXECUTIVE OFFICERS

FOR

1929-30

BIRGER THELE,
Vice Chairman.



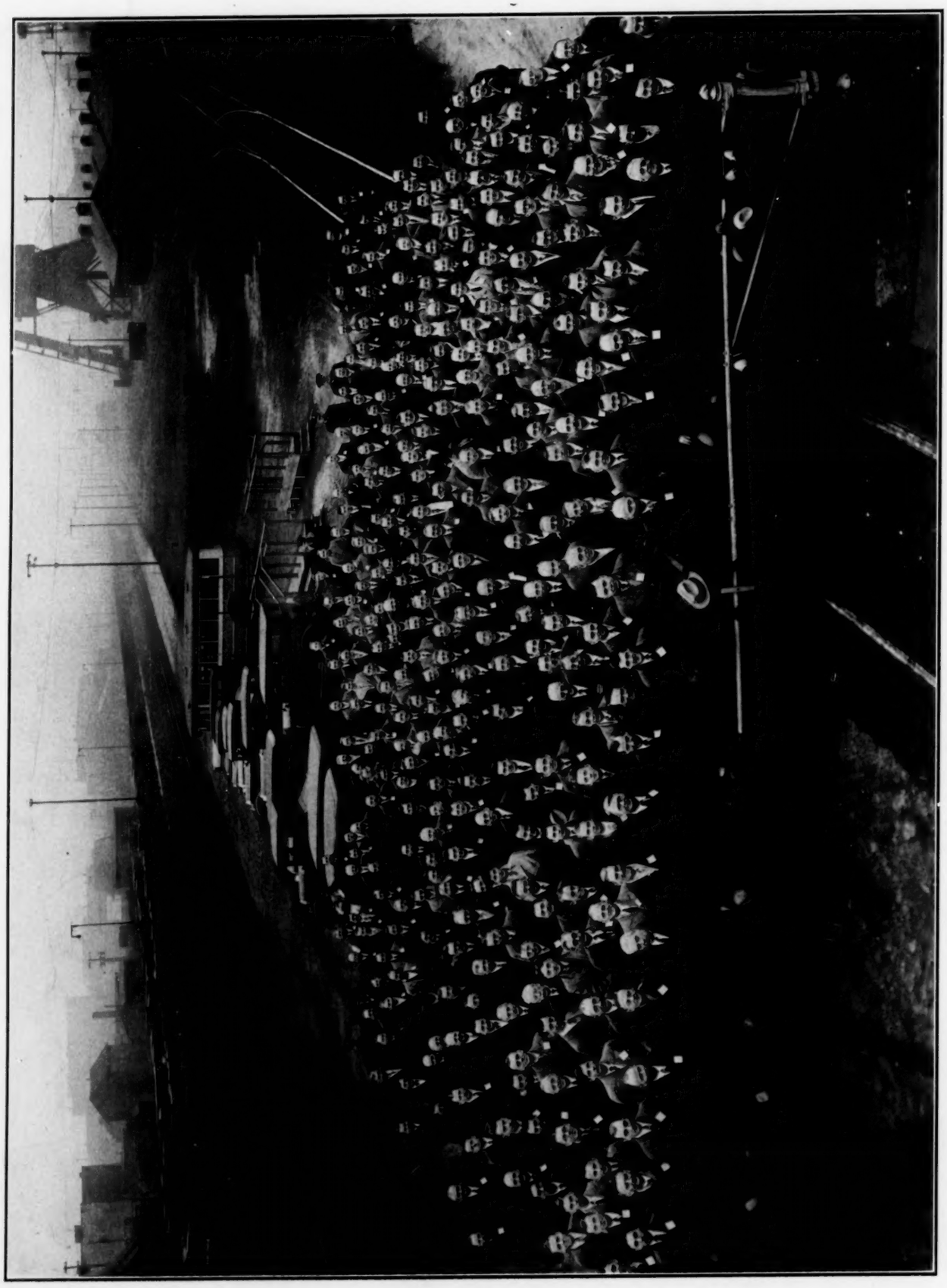
J. B. LOONEY,
Secretary and Treasurer.

E. C. WINFIELD,
Chairman.

The Birmingham District Section is one of the most progressive sections of the Association. It includes in its membership, most of the prominent Steel Mill Engineers in the Southern Territory. Meetings are held last Saturday of the month. The meetings are well attended and many valuable contributions to the Association Proceedings have come from this Section.

E. P. WINTERS,
Past Chairman, Member of Executive Committee.

J. E. SAYER,
Past Chairman, Member of Executive Committee.



Association of Iron and Steel Electrical Engineers and Cleveland Engineering Society, Inspection Trip to National Tube Company, Lorain, Ohio, December 6, 1929.